TRANSACTIONS

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

VOLUME 31

THIRTY-FIRST ANNUAL MEETING
BOSTON, JANUARY 27–30, 1925

SEMI-ANNUAL MEETING
ATLANTIC CITY, N. J., JUNE 15–17, 1925

PUBLISHED BY THE SOCIETY AT THE OFFICE OF THE SECRETARY
29 WEST THIRTY-NINTH STREET
NEW YORK, N. Y., U. S. A.

CONTENTS

Officers and Council

American Society of Heating and Ventilating Engineers

1925

President . . . . S. E. Dibble, Pittsburgh, Pa.
First Vice-President . . . . Wm. H. Driscoll, New York, N. Y.
Second Vice-President . . . . F. Paul Anderson, Pittsburgh, Pa.
Treasurer . . . . Perry West, Newark, N. J.
Secretary . . . . F. C. Houghten, New York, N. Y.

Council

S. E. Dibble, *Chairman*

Wm. H. Driscoll, *Vice-Chairman* — F. C. Houghten, *Secretary*

Homer Addams — W. T. Jones
F. Paul Anderson — Thornton Lewis
W. H. Carrier — J. H. Walker
J. A. Cutler — Perry West
W. E. Gillham — A. C. Willard

Advisory Council

Homer Addams, *Chairman;* Henry Adams, R. P. Bolton, H. P. Gant, John Gormly, John F. Hale, H. M. Hart, E. Vernon Hill, J. D. Hoffman, S. A. Jellett, D. D. Kimball, J. H. Kinealy, S. R. Lewis, J. I. Lyle, J. R. McColl, D. M. Quay, C. L. Riley, C. B. J. Snyder, F. R. Still, W. S. Timmis.

Committees of the Council

Executive: William H. Driscoll, *Chairman;* Homer Addams, W. H. Carrier.
Finance: Thornton Lewis, *Chairman;* J. A. Cutler, Perry West.
Membership: W. H. Carrier, *Chairman;* W. E. Gillham, W. T. Jones.
Publication: A. C. Willard, *Chairman;* F. Paul Anderson, J. H. Walker.

Research Department

Committee on Research: Wm. H. Driscoll, *Chairman;* O. P. Hood, *Member ex-officio;* F. Paul Anderson, *Director of Research Laboratory.* H. M. Hart, C. V. Haynes, J. I. Lyle, Perry West (3 years); Homer Addams, E. Vernon Hill, Alfred Kellogg, J. R. McColl, F. R. Still (2 years); S. E. Dibble, C. F. Eveleth, F. B. Howell, Thornton Lewis, W. S. Timmis (1 year).

Nominating Committee: H. P. Gant, *Chairman;* Alfred Kellogg, J. H. Kitchen, G. C. Morgan, M. F. Rather.

Guide Publication Committee: Perry West, *Chairman;* J. E. Bolling, C. E. Eveleth, E. S. Hallett, L. A. Harding, C. V. Haynes, F. D. Mensing, J. F. McIntire, L. C. Soule, A. C. Willard.

Committee on Chapter Relations: C. W. Farrar, *Chairman;* John Howatt, W. T. Jones, F. D. Mensing, J. H. Walker.

Committees—1925

Committee on Code of Ethics: J. R. McColl, *Chairman;* H. M. Hart, Alfred Kellogg, Henry C. Meyer, Jr., S. A. Jellett.

Committee on Code for Testing Low Pressure Heating Boilers: H. W. Brooks, *Chairman;* Homer Addams, F. Paul Anderson, John Blizard, C. E. Bronson, F. B. Howell, J. F. McIntire.

Committee on Chapter Suggestions: W. T. Jones, *Chairman;* C. W. Farrar, A. L. Sanford.

Committee on Legislation: W. G. R. Braemer, *Chairman;* F. F. Bahnson, C. R. Bradbury, S. A. Challman, Ralph T. Coe, F. I. Cooper, H. C. Eicher, H. H. Fielding, L. M. Frederick, S. R. Lewis, J. J. Mason, Wm. Mallis, H. M. Miller, J. G. Pease.

Committee on Increase of Membership: C. V. Haynes, *Chairman;* C. W. Farrar, F. J. Friedman, F. H. Gaylord, E. F. Glore, C. P. Hackett, H. P. Kahn, H. E. Pursell, E. A. Stark, E. S. Storm, M. F. Thomas, J. F. Tuttle, K. Wright.

Committee to Draft a Code for he Guidance of Membership Committees: Alfred Kellogg, *Chairman;* H. P. Gant, W. E. Gillham.

Committee on Revision of Constitution: E. V. Hill, *Chairman;* Homer Addams, R. P. Bolton, H. P. Gant, John Gormly, John F. Hale, H. M. Hart, J. D. Hoffman, S. A. Jellett, D. D. Kimball, J. H. Kinealy, S. R. Lewis, J. I. Lyle, J. R. McColl, D. M. Quay, C. L. Riley, C. B. J. Snyder, F. R. Still, W. S. Timmis.

Committee on Schoolhouse Standards: H. C. Eicher, *Chairman;* J. D. Cassell, F. I. Cooper, H. P. Dempsey, E. S. Hallett, John Howatt, S. R. Lewis.

Committee on Pipe Sizes: J. B. Walker, *Chairman.*
University Professors: J. E. Emswiler, F. E. Giesecke.
Contracting Engineers: W. L. Fleisher, H. M. Hart.
Consulting Engineers: R. V. Frost, W. R. Jones, W. S. Timmis.
Manufacturers: J. A. Donnelly, C. E. Eveleth, C. V. Haynes.
Central Station Heating: J. C. Hobbs, J. H. Walker.

Committee on Code of Heating & Ventilating: L. A. Harding, *General Chairman.*

Sub-Com. I.	Definition of Terms	F. Paul Anderson, *Chm.*
Sub-Com. II.	Ventilation Requirements for Public Buildnigs	E. Vernon Hill, *Chm.*
Sub-Com. III.	Requirements for Heating Buildings	A. C. Willard, *Chm.*
Sub-Com. IV.	Direct Steam or Hot-Water Radiation	R. V. Frost, *Chm.*
Sub-Com. V.	Indirect Steam or Hot-Water Radiation	L. C. Soule, *Chm.*
Sub-Com. VI.	Heating Boiler Capacity	J. F. McIntire, *Chm.*
Sub-Com. VII.	Warm-Air Furnace Heating	J. D. Hoffman, *Chm.*
Sub-Com. VIII.	Design of Chimneys and Flues	J. R. McColl, *Chm.*
Sub-Com. IX.	Pipe Sizes for Steam Heating	J. A. Donnelly, *Chm.*
Sub-Com. X.	Pipe Sizes for Hot-Water Heating	W. S. Timmis, *Chm.*
Sub-Com. XI.	Air Ducts for Ventilation	C. A. Booth, *Chm.*
Sub-Com. XII.	Air Washers and Humidifiers	W. H. Carrier, *Chm.*
Sub-Com. XIII.	Pumps for Heating Systems	Perry West, *Chm.*
Sub-Com. XIV.	Standard Symbols for Drawings	J. H. Walker, *Chm.*

Committee on Standards of Ventilation: W. H. Carrier, *Chairman;* F. R. Still, *Vice-Chairman;* E. P. Bradley, Philip Drinker, F. R. Ellis, E. S. Hallett, Thomas Chester, E. V. Hill, F. C. Houghten, John Howatt, J. R. McColl, R. R. Sayers, Perry West, A. C. Willard.

Committee to Confer with American Institute of Architects: Perry West, *Chairman;* D. K. Boyd, S. A. Jellett, D. D. Kimball, S. R. Lewis, J. R. McColl, C. L. Riley, F. H. Willis.

Committee to Confer with A. S. M. E. Boiler Code Committee: W. S. Timmis, *Chairman;* Homer Addams, F. Paul Anderson, L. P. Breckenridge, James Doherty, H. M. Hart, F. B. Howell, J. F. McIntire.

Committee to Confer with the International Exposition of Modern Decorative and Industrial Arts, Paris, France, June 1925: Henry S. Downe and Auguste Beaurrienne.

Committee to Develop a Ventilation Safety Code under the Procedure of the American Engineering Standards Committee: W. S. Timmis, *Chairman;* W. H. Carrier, E. V. Hill, F. C. Houghten, D. D. Kimball, J. R. McColl, R. R. Sayers, F. R. Still, A. C. Willard.

Officers of Local Chapters

1925

Cleveland

Headquarters, Cleveland

Meets: Second Friday in Month
President, T. A. Weager
Vice-President, M. J. Rather
Secretary, C. W. Colby
Treasurer, R. E. Bronson
Board of Governors: J. J. Kissick, J. J. Mason, Ernest Szekely

Colorado

Headquarters, Denver

Meets: Second Monday in Month
President, L. A. Michael
Vice-President, O. G. Ward
Secretary, F. E. Price
Treasurer, H. H. Fielding
Board of Governors: J. H. Daly

Illinois

Headquarters, Chicago

Meets: Second Monday in Month
President, Homer R. Linn
Vice-President, A. B. Martin
Secretary, H. G. Thomas
Treasurer, August Kehm
Board of Governors: J. A. Cutler, C. D. Allan, E. P. Heckel

Kansas City

Headquarters, Kansas City, Mo.

Meets: First Monday in Month
President, Nate W. Downes
Vice-President, B. F. Cook
Secretary, R. B. Johnson
Treasurer, B. C. Naylor
Board of Governors: W. N. McIntyre, C. H. Carr, David Caleb

Massachusetts

Headquarters, Boston

President, E. A. Dussosoit
Secretary, J. W. Brinton
Treasurer, W. T. Smallman
Board of Governors: D. S. Boyden, E. A. Dussosoit, W. T. Jones, Alfred Kellogg, R. E. Shaw, H. B. Wiegner, E. H. Whittemore

Michigan

Headquarters, Detroit

Meets: First Monday after the 10th of the Month
President, F. W. Johnson
Secretary, N. B. Hubbard
Treasurer, H. A. Hamlin
Board of Governors: N. J. Hill, L. L. Smith, E. R. Little

Minnesota

Headquarters, Minneapolis

Meets: Second Monday in Month
President, A. M. Wagner
Vice-President, R. W. Otto
Secretary-Treasurer, E. B. Gordon, Jr.
Board of Governors: A. L. Sanford, Albert Buenger

New York

Headquarters, New York

Meets: Third Monday in Month
President, A. A. Adler
Vice-President, R. H. Carpenter
Secretary, E. B. Johnson
Treasurer, F. E. W. Beebe
Board of Governors: W. J. Olvaney, Thomas J. O'Donnell, H. P. Kahn

Western New York

Headquarters, Buffalo

Meets: First Monday in Month
President, W. G. Fraser
First Vice-President, Herman Worsham
Second Vice-President, H. P. Dempsey
Secretary, Roswell Farnham
Treasurer, Calvin Wadley

Ontario

Headquarters, Toronto, Can.

Meets: First Monday in Month
President, H. R. Flett
Vice-President, A. J. Dickey
Secretary, M. W. Shears
Treasurer, H. J. Church
Board of Governors: H. H. Angus, A. S. Leitch, M. F. Thomas

Philadelphia

Headquarters, Philadelphia

Meets: Second Thursday in Month
President, R. P. Schoenijahn
Vice-President, Benj. Adams
Secretary, Frank R. Lord
Treasurer, J. V. Cavileer
Board of Governors: R. C. Bolsinger, C. F. Eveleth, B. W. Wilson

Pittsburgh

Headquarters, Pittsburgh

Meets: Second Monday in Month
President, J. E. McGinness
Vice-President, T. M. Dugan
Secretary, Margaret Ingels
Treasurer, Lee Phillips
Board of Governors: J. B. Walker, E. C. Evans, S. E. Dibble

St. Louis

Headquarters, St. Louis

Meets: Third Thursday in Month
President, A. M. Lane
First Vice-President, E. B. Langenberg
Second Vice-President, J. N. Foster
Secretary, Thomas J. C. Gale
Treasurer, E. H. Quentin
Board of Governors: D. M. Forgan, T. V. Hester, W. C. Sodeman, R. K. Milward

Wisconsin

Headquarters, Milwaukee

Meets: Third Tuesday in Month
President, R. G. Olson
Vice-President, H. W. Page
Secretary, F. R. Dannies
Treasurer, W. H. Wilson
Board of Governors: W. F. Noll, C. W. Miller, G. L. Larson

TRANSACTIONS

of

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

No. 710

THE THIRTY-FIRST ANNUAL MEETING, 1925

QUESTIONS of vital import to every member of the Society were decided at its 31st Annual Meeting January 27–30, with a decisiveness that left no doubt as to the feeling of the membership regarding the desire to sponsor research by an increase in dues, to continue the splendid code work and in every way fulfill the ideals of its founders in service to mankind.

In the atmosphere and on the very ground where so many of the momentous decisions were made that affected the world's history, it was fitting that the Society should act in the manner of its forefathers and adopt an ambitious program for future activities.

There could be no question about the brightness of the future and enthusiastic support of the actions taken, when it was remembered how spontaneous was the response to an appeal to support the research program.

Society affairs generally were in excellent shape and it was evident from the reports of the Officers and the Committees in charge of the varied activities that a fruitful and profitable year had been passed and that the future could be faced confidently. The professional program had a great appeal and every session had a good attendance.

Boston can well be proud of its first meeting. It not only was big but it made history. The 350 who were present will testify to that.

Pres. Homer Addams declared the 31st Annual Meeting of the Society in session for the transaction of business at 10:30 A.M., January 27 in the Engineering Societies' Building, New York. The reports of the Officer of the Council andC ommittees were diverted to the Bostion sessions of the meeting.

W. H. Driscoll brought up the matter of the Amendment to the By-Laws stating that an unofficial ballot indicated that a majority of the members favored it and suggested that before formal action was taken that it be discussed at the sessions in Boston and that the meeting adjourn until Saturday morning, January 31, for the official ballot.

Chairman Frank G. McCann presented the report of the tellers of election of officers, and the business session was then adjourned until January 31.

The first Boston session opened on Wednesday morning, January 28, at 10 o'clock, with Pres. Homer Addams presiding. Members and guests were welcomed to Boston by Alfred Kellogg, general chairman of the Annual Meeting, and by Mark Sullivan, corporation counsel of the City, acting in behalf of the Mayor.

The reports of the President, Council, and Secretary were then called for and read. These were followed by the presentation of the first professional paper of the Meeting.

The third session was given over to the discussion of the Code of Minimum Requirements for Heating and Ventilation of Buildings, General Chairman L. A. Harding presiding. President Addams appointed a committee to consider what form the Code should take and to make recommendations at a later date. Sections 6, 10, and 11 were discussed and approved.

Reports of the Membership Committee and the Committee on Increase of Membership were presented by C. V. Haynes at the opening of the fourth session. A resolution, proposed by A. M. Lane of St. Louis suggesting a plan for more speedy election of members to the Society was unanimously adopted.

Reports of the Publication Committee and the Finance Committee were next read. In a few preliminary remarks, Thornton Lewis explained that it was a pleasure to bring his report before the Society and with the aid of lantern slides he showed the balance sheet and budget comparisons for the past two years and commented on the financial showing made during the year. Summarizing he said that during the year 1924, the business of the Society increased 20 per cent. The excess of income over ordinary Society expenses was 32 per cent. While the increase of net worth was 50 per cent. Cash on hand increased 63 per cent, while the increase in the gross profit of publications was 73 per cent and in the net profit of this department 1600 per cent.

Following the report of the Guide Publication Committee the session was devoted to the presentation of professional papers.

The Research Session opened with the reading of several papers and was then turned over to W. H. Driscoll, chairman of the Committee on Research. He gave the following report of the activities of the Laboratory:

Report of Committee on Research

It seems eminently fitting, at a time when we have before us the serious problem that confronts us today, that we should be meeting in this historic section where so many men and women of an earlier day offered up their hearts, their homes, and their lives in support of an ideal.

The men of Lexington and Concord, the embattled farmers of Bunker Hill, Paul Revere, and the other heroes of Revolutionary times, were men of vision and imagination, of courage and determination. They set aside their personal interests to devote themselves to the promotion of an idea that meant hardship and struggles for them, and offered no immediate return beyond the eternal hope that has ever filled the hearts of men and has been the propelling force that has carried civilization forward. Their dream was of a nation founded on liberty and justice and one in which individual thought and action would have its freest expression.

In their most fanciful thoughts, they could hardly have imagined that the "dreamland" which they were struggling to establish would in strength, power and influence, take its place in the front rank of the nations of the world. Nor is it likely that they

would have struggled on against almost overwhelming odds, in spite of misfortune and disappointment to a final success, had they for one moment assumed that the nation they were struggling to establish would soon forget their sufferings and their struggles and concern itself so completely with the material things of life as to leave no room for sentimentalism, for altruism, or for idealism.

Has the vision of this great organization, calling itself "The American Society," become so warped that it can see nothing but its own selfish interests? Have its ideals fallen to such a low plane that it has no room for the finer sentiments of life? Have its ideas become so material that it must demand the immediate delivery of a pound of flesh for every dollar it invests?

If so, hasn't it forgotten it's Americanism? Hasn't it forgotten America's great heroes and hasn't it forgotten the very first object for which it was organized, *viz.*, the promotion of the arts and sciences connected with Heating and Ventilation in all its branches.

The members of our Society owe their positions in the world today largely to their knowledge of the science of heating and ventilating. They exchange that knowledge for a share of the nation's wealth with scarcely any realization of the fact that the knowledge they have is not an individual possession, but is a heritage that they have acquired from their predecessors, just as the great privilege of citizenship in this nation is a priceless heritage the value of which has been developed through succeeding generations with little or no effort on their part.

There is a moral obligation, however, that goes with their citizenship that men are often inclined to forget, and that is the obligation of developing its value to the fullest extent, so that the nation of which they are a part will maintain its place in the world.

This obligation does not come and go periodically. It is ever present with us and is not fulfilled by wearing patriotic emblems, singing patriotic selections, or publicly flaunting the nation's flag. These are not the essentials of patriotism, but frequently are evidences of hysteria. Nor are our obligations fully discharged when, in times of danger, we rally to the defense of the flag.

We are failing to appreciate the great value of citizenship to which we have fallen heir, and are very deficient in patriotism, when we fail to make use of the means at our disposal to develop and increase the value of that citizenship so that it may be passed on as a worthy heritage to our successors.

There is no better method of accomplishing this purpose than by increasing and spreading the knowledge of the subjects in which we are better versed than our fellow-citizens.

Individually, we may study and analyze and ponder over these problems and endeavor to develop special knowledge of the subject, but few of us are so constituted mentally as to be able to offer much to the world through this method.

The spirit of interest in research work is given to but few men, and it is from the application of minds that are urged on by this spirit that results are obtained that are of real benefit to mankind, and that development and progress in the scientific world is accomplished.

Why should we not, then, devote our time, as most of us are doing, to the things in which we are most efficient and, from the personal profits that come from such activities, contribute our mite to the support of an institution devoted to this highly specialized work.

It will be largely through the result of scientific research that our nation will maintain its place in the world, and the support of this work is one of the essentials of patriotism.

There has never been offered a better opportunity for those of us who are interested in the problems of heating and ventilating to display our appreciation of what our citizenship means than by a whole-hearted support of the Research Laboratory.

This support should be forthcoming even though we may differ in some respects with its methods of operation and have some dissatisfaction with the results it is accomplishing.

It is a human institution and it is far from perfect but, in the distance it will arrive at that state of perfection which is visualized in the minds of those who have struggled so hard to establish and maintain it. That distance will be increased or diminished in

proportion to the moral and financial support which the members of our society offer and to the extent to which they interest themselves in its activities.

The Research Laboratory was established and its income solicited on a five years' preliminary basis, with the thought in mind that, by the time we had been through the experiences of those five years, we would have discovered the value, or lack of value, of such an institution. If the institution lacked value, it could not hope to continue. If it had a value, it was hoped that some definite and substantial scheme of financing it would be evolved, so that its existence and continuance would be definite and certain.

In the face of the evidence that we have as to the great work the Laboratory has accomplished, is there any man in the Society who will dare arise and say that there is no justification for its existence?

Take the investigations in connection with the matter of effective temperature and the establishment of comfort zones. The value of this work has received the widest recognition and commendation from such men as Dr. Leonard Hill and Dr. Vernon of England who are accepted as the world's authorities on this subject.

It is also highly commended by the Harvard Medical School, U. S. Public Health Service, and other institutions in this country interested in the subject of good health as affected by atmospheric or ventilation conditions. Various reports on this investigation published in the JOURNAL are widely sought after and have resulted in our JOURNAL receiving recognition in many Medical and Public Health Libraries.

The data contained in these reports and published in THE GUIDE give fairly complete and accurate information of great value to engineers in designing systems for increasing human comfort, either by cooling, humidifying, dehumidifying, or by air motion.

From the data now available, an engineer can pre-determine just what effect a change in atmospheric conditions, as pertaining to humidity, or air motion, will have on human comfort.

The Direct Radiation Tables first published in the December 1921, JOURNAL and now appearing in THE GUIDE are a direct contribution from the Research Laboratory. They are of inestimable value and are in constant use in the offices of a great many heating engineers.

The work on air leakage and infiltration is still very incomplete, yet is widely sought after by architects and engineers interested in determining heat lost from buildings. The data published to date has been considered of sufficient value to be reprinted in a number of Trade Journals interested in heating and ventilation, as well as Journals of interest to Architects.

In addition to the great work that has been done at the Laboratory, what have been its effects on the Society?

When the Laboratory was established we had less than nine hundred members—today we have close to two thousand; we had eight chapters—today we have fourteen.

We know, of course, that we have among our members some who are not willing to give this institution its proper support. We know, too, unfortunately, that we have some whose circumstances will not permit them to do so.

Both of these classes we will probably lose if they are forced to contribute to the Laboratory's support. This is unfortunate, but this great movement cannot be abandoned because of minor obstacles and the loss of these members must be weighed against the loss of those who will terminate their membership if the Laboratory ceases its existence.

In the final analysis, which class of member can we least afford to lose?

How much money will it take to run the Laboratory?

Our contract with the Bureau of Mines obligates us to spend a minimum of Fifteen Thousand Dollars per annum, in consideration of which the Bureau furnishes us with space, facilities, equipment, and service, that we could not obtain for three times that amount elsewhere.

We must have a Director, and we must have engineers and assistants. Research work is personal, we do not produce a machine-made product, therefore, the extent of our activities depends on our ability first, to obtain the type of men and women required; second, to obtain the income with which to employ them.

We have spent as high as $33,000 per year. Last year we reduced this to $23,000. The last six months of the year we were operating on a basis of about $1500 per month. This was done by curtailing our activities to a minimum, and by reducing our organiza-

tion and by accepting, without compensation, the services of Dean Anderson which during that period he generously contributed to us.

A tentative budget for 1925 indicates that we ought to spend about $25,000.

As previously explained, if we are to merely exist, we must spend $15,000 but, in so doing, we would practically be defaulting in our contract with the Bureau of Mines, which at least implies that the results which we obtain will justify the continuance of our relations with them. We cannot continue on the hand-to-mouth existence that characterizes our methods at the present time. We cannot carry out satisfactory programs with a condition of bankruptcy constantly staring us in the face. We must have a basic income which will at least insure our continued existence. Beyond that, we may expand as our resources permit.

On the basis of our present membership with all allowances for contingencies, an increase of $10.00 per year for full and associate members would net us at least $15,000. With this as a basis we could operate, and we would still have the income from THE GUIDE to fall back on.

Many members are eager and willing to contribute beyond the bare necessities, and we could further appeal to the manufacturing interests, with the assurance that they would at least not have the excuse that we were not carrying our share of the burden.

With such a financial arrangement, the members of your Committee on Research could devote themselves to the consideration of the subjects to be handled, to discussions with the membership of the work, and to the arrangement of a program that would conform to the reasonable desires of members and contributors.

Under the present scheme of operation they are so concerned with the problem of obtaining funds that they have little or no time for anything else.

If there is anything radically wrong with the Laboratory, that is the answer.

Dean Anderson, Director of the Research Laboratory followed the report of the Research Committee with an address on the influence and value of research work.

The next matter brought up for discussion was the proposed amendment to the constitution.

AMENDMENT TO THE BY-LAWS OF THE CONSTITUTION

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

Members of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS,

In accordance with the resolution passed last June at the 1924 Semi-Annual Meeting of the Society,

> "RESOLVED, that an Amendment to the By-Laws of the Constitution be drawn up providing for an increase of $10.00 in the dues of Members and Associate Members, said increase in dues to be deposited in the Research Fund and to be used for no other purpose, and that the Secretary be guided by the present Constitution and By-Laws in submitting the proposed amendment to the Membership in order that it may be voted upon at the next Annual Meeting of the Society and become effective January 1, 1925."

and in accordance with ARTICLE XIII of the By-Laws of the Constitution of the Society:

> "ARTICLE XIII, *Section 1.* The By-Laws of the Society shall be subject to addition, amendment or repeal by a majority of the members present and voting at any Annual Meeting of the Society, or at any Special Meeting of the members called for that purpose, provided that a copy of such proposed addition, amendment or repeal shall have been submitted in writing to all of the members entitled to vote, at least thirty (30) days prior to the time of the meeting at which such addition, amendment or repeal is to be considered."

there is hereby submitted a copy of a Proposed Amendment to Sections 2, 3 and 5 of Article III of the By-Laws. This Amendment *will be voted upon* at the Annual Meeting of the Society in January, and if adopted will become effective January 1, 1925.

F. C. HOUGHTEN, *Secretary.*

ARTICLE III, SECTIONS 2, 3, AND 5 OF THE BY-LAWS—"AS NOW IS"—

Section 2. The annual dues of Members and Associate Members shall be fifteen dollars ($15.00), of Junior Members ten dollars ($10.00) and of Student Members five dollars ($5.00).

Section 3. Of the annual dues paid by the members of each grade, three dollars ($3.00) shall be considered as a subscription paid for the JOURNAL.

Section 5. The dues of a new member of any grade except a Junior Member may be pro-rated monthly for the balance of the year, but if the amount thus paid is less than five dollars ($5.00) such member shall not be entitled to receive the volume of the TRANSACTIONS for the year in which he is elected, but he shall otherwise be entitled to all the rights and privileges of membership.

ARTICLE III, SECTIONS 2, 3 AND 5 OF THE BY-LAWS—"AS PROPOSED TO BE AMENDED"—

Section 2. The annual dues of Members and Associate Members *shall be Twenty-five Dollars ($25.00)*, of Junior Members ten dollars ($10.00) and of Student Members five dollars ($5.00).

Section 3. Of the annual dues paid by the members of each grade, three dollars ($3.00) shall be considered as a subscription paid for the JOURNAL *and the annual dues paid by Members and Associate Members ten dollars ($10.00) shall be considered as a direct contribution to the Research Fund and shall be immediately deposited in said fund and shall not be used for any other purpose.*

Section 5. The dues of a new member of any grade except a Junior Member may be pro-rated monthly for the balance of the year but if the amount thus paid is less than five dollars ($5.00) such member shall not be entitled to receive the volume of the TRANSACTIONS for the year in which he is elected, but he shall otherwise be entitled to all the rights and privileges of membership. *Forty per cent (40%) of the pro-rated dues of Members and Associate Members shall be considered as a contribution to the Research Fund and shall be immediately deposited in such fund and shall not be used for any other purpose.*

The action of the meeting on Tuesday morning in New York was reviewed and it was announced that the informal vote was 413 in favor and 301 against. After a discussion in which the views of the Minnesota, St. Louis, Massachusetts, New York, Illinois and Western New York Chapters were presented, as well as the ideas of a great many individual members, the motion that "it is the sense of this meeting that we recommend to the meeting to be held in New York on Saturday, January 31, that this amendment be adopted," was voted upon and carried unanimously.

E. B. Langenberg presented two resolutions referring to Chapter representation and methods of amending the By-Laws of the Society which were unanimously adopted.

The final session of the Meeting at Boston was devoted to professional papers and the installations of officers.

The adjourned business session was opened at Headquarters Office in the Engineering Societies Building in New York, with Homer Addams presiding. After the announcement of a quorum, the proposed amendment to the Constitution and By-Laws was read, discussed and finally adopted. Whereupon the meeting adjourned.

Registration, housing and entertainment of the 350 members and guests were handled perfectly by the chapter committee, headed by Alfred Kellogg, general chairman.

On the day of the arrival of the guests, an informal reception was held at 6:30 P.M. and a special "get-acquainted" dinner was served to the officers, members of the council, and their wives by the Boston Committee.

On Wednesday afternoon a tea and bridge party was given for the ladies and on Wednesday evening, preceding the professional session, the past presidents met for dinner.

Thursday afternoon was given over to visiting historical spots and industrial plants around Boston.

The most elaborate social event of the meeting was the annual banquet and dance

held in the ball room of the Copley-Plaza Hotel. At the conclusion of the banquet, Alfred Kellogg, toastmaster, presented to President Addams a diamond studded watch fob as an emblem of the Society's appreciation of his long and untiring service.

PROGRAM ANNUAL MEETING 1925

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

All Professional Sessions in Copley-Plaza, Boston, Mass.

First Session

Tuesday, January 27, 10 a. m.

Engineering Societies Building, New York

BUSINESS SESSION

Announcement of Quorum
Appointment of Tellers of Annual Election
Address of President
Report of Council
Report of Secretary
Report of Treasurer
Report of Committee
- *a.* Executive
- *b.* Finance
- *c.* Publication
- *d.* Membership

Report of Tellers of Election
New Business

Adjournment to Boston (Leave New York for Boston at 12 o'clock noon and arrive Boston 5 P.M.)

Second Session

Wednesday, January 28, 10 a. m.

PROFESSIONAL SESSION

Address of Welcome, Mayor Curley, Boston
President's Address
Response by Alfred Kellogg

Paper:
Heating and Ventilating the Modern Steam Power Station, by Davis S. Boyden and A. B. Williams

Paper:
Effect of Temperature upon the Friction of Water in Pipes, by F. E. Giesecke

Third Session

Wednesday, January 28, 8 p. m.

PROFESSIONAL SESSION

Discussion of Code of Minimum Requirements for the Heating and Ventilation of Buildings, by L. A. Harding, General Chairman

Fourth Session

Wednesday, January 28

PROFESSIONAL SESSION

8:00 P.M.
Resumé of Business Session in New York—Report of Technical Committees

Paper:
Industrial Application of Ozone, by F. E. Hartman

Fifth Session

Thursday, January 29, 10 a. m.

RESEARCH SESSION

Report of Committee on Research, by Wm. H. Driscoll, Chairman of Research Committee

The Influence of Research, F. Paul Anderson, Director of Research Laboratory, A.S.H.&V.E.

Paper:
The A-A Dust Determinator, by Margaret Ingels

Paper:
The Use of Owen's Jet Dust Counter and of Electric Precipitation on the Determination of Dusts, Fumes and Smokes in the Air, by Philip Drinker

Paper:
Further Data on Infiltration of Air through Building Openings, by C. C. Schrader

Paper:
Equivalent Conditions of Temperature, Humidity and Air Movement Determined with Individuals Normally Clothed. "Effective Temperature with Clothing," by C. P. Yaglou and W. Edw. Miller

Paper:
Basal Metabolism, Before and After Exposure to High Temperature and Humidities, by Dr. W. J. McConnell and C. P. Yagloglou

Paper:
Work Tests Conducted in Atmospheres of High Temperatures and Various Humidities, in Still and Moving Air, by Dr. W. J. McConnell, C. P. Yagloglou, assisted by W. Edw. Miller and W. B Fulton

Paper:
A Progress Report of the Critical Velocity of Steam Flow in Two Pipe Systems. Work done by Louis Ebin and Gordon H. Eisenhart. Reported by Margaret Ingels

Thursday, January 29, 2 p. m.

Inspection Trips: U. S. Cartridge Co., Lowell; B. F. Sturtevant Co., Hyde Park; Weymouth Plant Edison Elect. Ill. Co.

Sixth Session

Friday January 30, 10 a. m.

PROFESSIONAL SESSION

Paper:
Basing Warm Air Heating Selections on Climatological Conditions and Heater Performance Curves, by V. S. Day

Paper:
Some Facts about Enclosed Radiation, by R. V. Frost

Paper:
Characteristics of an Air-Tube Heater, by L'Roche, G. Bousquet and George A. Foisy

Seventh Session

Friday, January 30, 2 p. m.

PROFESSIONAL SESSION

Paper:
Heat Transference and Combustion Tests in Small Domestic Boilers, by H. W. Brooks, N. L. Orr, W. N. Myler, Jr. and C. A. Herbert

Paper:
Designing and Planning for Home Heating Economies with Especial Reference to Anthracite Utilization, by D. Knickerbacker Boyd

Paper:
Small Size Anthracite Coal, by C. A. Connell

No. 711

EFFECT OF TEMPERATURE UPON THE FRICTION OF WATER IN PIPES

By F. E. Giesecke, Austin, Texas

MEMBER

IN CALCULATIONS relating to water heating, especially for forced circulation systems, it is important to know how the friction varies with the temperature of the water.

Since the effect of temperature upon the friction of water is relatively small, very little attention has been given to this subject by investigators, particularly by those who were interested in securing data for application to general hydraulic calculations rather than to problems arising in the heating of buildings by the circulation of hot water. In several instances important research has been conducted to determine the friction of water without even recording the temperature of the water employed in the experimental determinations.

It is now quite generally understood that the friction of water in pipes may be determined by the formula

$$h = f \frac{l}{d} \frac{v^2}{2g} \qquad (1)$$

where h = friction head, in feet of water,
l = length of pipe, in feet,
d = diameter of pipe, in feet,
v = velocity of water, in feet per second,
g = acceleration due to gravity, in feet per second each second,
f = dimensionless factor, whose value depends on the character of the pipe, the velocity of the water, the diameter of the pipe, the density of the water, and the viscosity of the water.

It has, so far, been impossible to describe the character of the pipe surface in mathematical terms; for that reason any one discussion of this formula should be limited to one particular kind of pipe, for example, to commercial black iron pipe, or to commercial galvanized iron pipe, or to smooth brass tubing, etc. In that event the value of f depends only on the diameter of the pipe and on the velocity, density, and viscosity of the water.

In ordinary hydraulic calculations the range of temperature of the water is small

---

Presented at the Annual Meeting of the American Society of Heating and Ventilating Engineers, Boston, January, 1925.

and hence the density and viscosity of the water are practically constant and f may be assumed to vary only with the diameter of the pipe and with the velocity of the water. For that reason, textbooks on hydraulics give tabular values of f based only on the diameter of the pipe and the velocity of the water.

When the range of temperature is so great that there is an appreciable change in the density and viscosity of the water, these two factors must also be considered and the four factors, which determine the value of f, must be combined in such a manner that the combination is dimensionless.

The four factors are expressed in the following units:

velocity, in feet per second (ft. sec.$^{-1}$)
diameter, in feet (ft.)
viscosity, in pounds per foot and second (lb. ft.$^{-1}$ sec.$^{-1}$)
density, in pounds per cu. ft. (lb. ft.$^{-3}$)

These four quantities can be combined in a number of different ways, but only the following combination will result in a dimensionless quantity:

$$\frac{\text{velocity} \times \text{diameter} \times \text{density}}{\text{viscosity}} = \frac{\text{ft. sec.}^{-1}\text{ ft. lb. ft.}^{-3}}{\text{lb. ft.}^{-1}\text{ sec.}^{-1}}$$

Since the density and viscosity of the water vary simultaneously with the temperature of the water, the two factors may be combined into one.

Prof. Osborne Reynolds suggested the name "kinematical viscosity" for the ratio of viscosity to the density. Following this suggestion we have:

$$\text{Kinematical viscosity} = \frac{\text{viscosity}}{\text{density}} = \frac{\text{lb. ft.}^{-1}\text{ sec.}^{-1} = \text{ft.}^{2}\text{ sec.}^{-1}}{\text{lb. ft.}^{-3}}$$

The expression $\dfrac{\text{velocity} \times \text{diameter}}{\text{kinematical viscosity}}$ is known as Reynold's number and will be so designated in this paper.

The character of the relation of f to Reynold's number has not yet been determined and until it shall have been, recourse must be had to experimental determinations.

It is possible that this relation may be given by some expression like

$$f = k\left(\frac{vd}{kv}\right)^{n}$$

where k and n depend on the character of the pipe and probably also on the diameter of the pipe and on the kind and character of the fluid in the pipe: *i. e.*, whether the fluid be water, oil, steam, air, etc., and whether it be hot or cold, compressed or dilated, etc.

Prof. Blasius probably made the earliest determination of this relation and published his findings, 1913, in *Mittheilungen uber Forschungsarbeiten, Vol. 131*, from which the dotted line shown in Fig. 1 was reproduced. This iine shows the relation between the friction factor, f, and Reynold's number for smooth brass pipe, based upon experimental determinations by Profs. Saph and Schoder, at Cornell University, published in *Tr. A. S. C. E.*, 1903.

It appears from Dr. Blasius' drawing that the brass tubing experimented with by Saph and Schoder and used by him in his investigation ranged in diameter from 1 in. to 2 in.

Prof. Blasius also studied the experimental determinations by Saph and Schoder of the friction factor for commercial galvanized iron pipe, but found so great a variation in these results that he concluded there was no definite relation between f and Reynold's number for other than very smooth pipes.

The writer made a study of the experimental determinations by Dr. Brabbee and published in *Gesundheitsingenieur*, 1913, and of the experimental results secured under his directions and published in University of Texas Bulletin No. 1759. Dr. Brabbee's tests were made with German commercial black iron pipe and covered a range of diameter from $^3/_4$ in. to $1^1/_2$ in. of velocity from 0.7 to 10 ft. per sec. and of temperature from 60 to 200 deg. fahr. The writer's tests were conducted with American commercial black iron pipe and covered a range of diameter from $^1/_2$ in. to 3 in., of velocity from 0.05 to 3 ft. per sec., and of temperature from 64 to 72 deg. fahr.

A study of these two series of tests shows quite clearly that the relation of f to Reynold's number depends on the diameter of the pipe so that, for a given Rey-

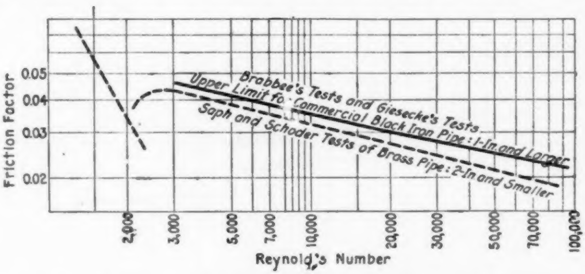


FIG. 1. RELATION OF FRICTION FACTOR TO REYNOLD'S NUMBER

nold's number the value of f decreases as the diameter of the pipe increases. The reason for this seems quite evident. The inner surface of a commercial black iron pipe has a certain degree of roughness and the degree of roughness is probably about the same for a 1-in. pipe as for a 10-in. pipe. Relatively, a given degree of roughness has a greater effect upon the friction of water flowing in the 1-in. pipe than on the friction of water flowing in the 10-in. pipe because the volume of water flowing per unit area of pipe surface is ten times as great for the 10-in. pipe as it is for the 1-in. pipe.

It seems then that for a given Reynold's number, the friction factor for a 10-in. pipe should be smaller than for a 1-in. pipe; in other words, a 10-in. pipe is, relatively, more nearly "perfectly smooth" than a 1-in. pipe.

This view is corroborated by a study of the friction factors published in Fanning's Treatise on Hydraulics and Water Supply Engineering, fifteenth edition, pp. 242–245, as shown in Fig. 2.

In preparing this figure it was assumed that Fanning's values apply at some average temperature, and 70 deg. was selected as the temperature for which Reynold's numbers were calculated. Velocities were selected, varying from 0.2 to 4 ft. per sec., and diameters, varying from 1 to 36 in. The diagram shows clearly that, if Fanning's values are correct, the friction factor decreases as the pipe diameter increases, for a given Reynold's number.

Since this conclusion differs materially from that arrived at by the authors of a paper on Flow of Fluids through Commercial Pipe Lines, published in *Engineering News-Record*, Oct. 26, 1922, it seems well to call attention to the discrepancy in order that additional research may be conducted, if necessary.

According to the writer's views, it is impossible to draw a single curve in Fig. 1 from which the friction factor for the entire range of pipe diameters can be determined and, therefore, only one curve is shown, the full line, from which the friction factor for 1-in. commercial black iron pipe can be found according to experimental determinations by Dr. Brabbee and by the writer. For larger pipe diameters, corresponding curves should be drawn below the one shown; how far below, the writer does not know.

If it were certain that the dotted line shown in the figure, applied to all smooth brass pipes, no matter how large the diameter, we could conclude that no curve giving the friction factor for commercial black pipe could lie below the dotted line and we could assume (in the absence of definite information) that a certain large

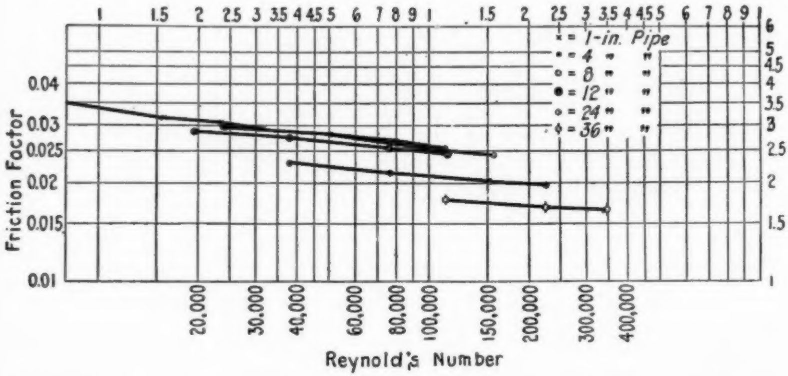


FIG. 2. RELATION OF FRICTION FACTOR TO REYNOLD'S NUMBER FOR FANNING'S VALUES OF THE FRICTION FACTOR

pipe, a 24-in. pipe, for example, could be considered "smooth" and that its curve should coincide with the dotted line of Fig. 1 and that, as the pipe diameter is gradually increased from 1-in. to 24-in., the corresponding curve is gradually moved from the full line of Fig. 1 to the dotted line of that figure. However, it is entirely possible, if careful tests were made with very large smooth brass pipe, that a curve may be found giving lower values of f than those shown by the dotted line of Fig. 1, and, consequently, it is also possible that lines showing the values of f for large commercial black iron pipe may lie below the dotted line of Fig. 1.

In order to apply the data available at the present time, to determine the effect of temperature upon the friction of water in pipes, the writer prepared Table 1 and Fig. 3, giving the kinematical viscosity of water at various temperatures. These values were deduced from the values of the densities and viscosities of water as given in the Landolt-Bornstein tables.

To explain the application of the data to the solution of practical problems, let it be required to find the friction head in a 1-in. black iron pipe, when the water is

flowing with a velocity of 2 ft. per sec. and when the temperature is (*a*) 60 deg. fahr. or (*b*) 180 deg. fahr.

The diameter of a 1-in. pipe is 1.049 inches or 0.0874 ft. Reynold's numbers for the two cases are:

$$\left(\frac{vxd}{kv}\right)$$

$$\frac{2 \times 0.0874}{0.00001207} \text{ or } 14{,}500 \text{ and } \frac{2 \times 0.0874}{0.00000362} \text{ or } 48{,}300$$

and the corresponding values of f (from Fig. 1) are 0.032 and 0.025.

The corresponding friction heads are $\left(\text{from } h = f\frac{1}{d}\frac{v^2}{2g}\right)$ 0.023 ft. of 60 deg. water and 0.018 ft. of 180 deg. water, per foot of pipe.

For some calculations it is best to express the friction head as a loss of pressure in pounds per square inch. To do this, in the above problem, we find, from a table

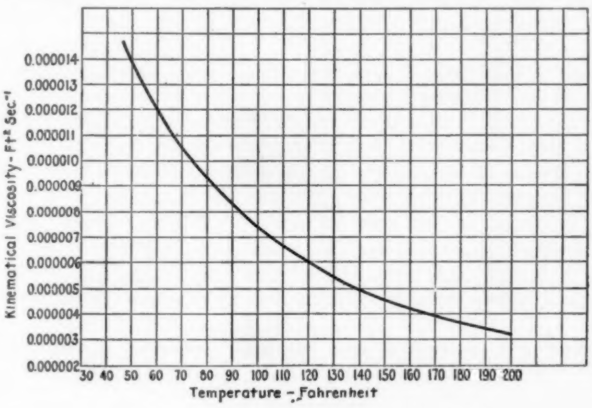


FIG. 3. THE KINEMATICAL VISCOSITY OF WATER

of thermal properties of water, the density of 60 deg. water to be 62.37 and that of 180 deg. water to be 60.58 lb. per cu. ft. The two friction heads correspond therefore, to losses of pressure of 0.01 and 0.0076 lb. per sq. in. The friction head for the 180 deg. water is, consequently, about 24 per cent less than that for the 60 deg. water, if the velocity is 2 ft. per sec. in both cases. In comparing these two friction heads it must be noted that a larger weight of water is conveyed when the temperature is 60 deg. than when the temperature is 180 deg., the velocity being the same in both cases. Since the quantity of heat conveyed by a column of water flowing in a pipe depends on the density as well as on the temperature of the water, it is important to consider the variation in velocity with the variation in density, when a constant mass of water is flowing through the pipe. Thus, in the above problem, let it be required to find the two friction heads when the 60 deg. water is flowing with a velocity of 2 ft. per sec. and the 180 deg. water with such a velocity that equal masses of water are transported through the two pipes in equal

units of time. In this case, the velocity of the 180 deg. water must be $\frac{2 \times 62.37}{60.58}$ or 2.06 ft./sec. The corresponding Reynold's number will be $\frac{48{,}300 \times 2.06}{2}$ or 49,700. The corresponding value of f will be practically the same as before, 0.025, but the corresponding friction head will be increased to $\frac{0.018 \times 2.06 \times 2.06}{2 \times 2}$ or to 0.019 ft. of 180 deg. water, which is still about 23 per cent less than that for the 60 deg. water.

If it is required to find the effect of temperature upon the friction of water flowing in a pipe larger than 1-in. it is necessary, in the absence of more definite information, to assume that the full line shown in Fig. 1 coincides with or is practically parallel to the line which represents the relation of f to Reynold's number for the particular pipe under consideration and to proceed as explained above for the 1-in. pipe.

TABLE 1. THE KINEMATICAL VISCOSITY OF WATER AT VARIOUS TEMPERATURES*

Temperature Deg. Fahr.	Kin. Visc. $Ft.^2$ $Sec.^{-1}$
50	0.00001397
60	0.00001207
70	0.00001054
80	0.00000928
90	0.00000823
100	0.00000735
110	0.00000660
120	0.00000596
130	0.00000494
150	0.00000454
160	0.00000419
170	0.00000389
180	0.00000362
190	0.00000337
200	0.00000312

* Translated and compiled by the writer from tables of densities and viscosities published, Landolt-Bornstein, Physikalisch Chemische Tabellen.

DISCUSSION

H. M. HART: If these experiments have only been tried out in a 1-in. pipe, I would like to ask how we are going to write a code on the total installation of a heating system which this Society is willing to have enacted into a law? I am particularly interested in this question of sizing of pipe for hot water heating. I have tried out a great many different systems of hot water heating and different tables of pipe sizing and I have gotten into trouble a good many times. It is probably due to my slowness in interpreting the rules that have been laid down by the particular author of the table I have been trying to follow, but I would like to guard against publishing something in a code that is drawn too fine, too close to the line. I like a good big factor of safety.

I know that this research work is necessary and it has to be worked out very accurately and we want the real results of this research work, but when we apply it to practical tables, then I think we need liberal factors of safety, because, in designing heating installations, it is next to impossible to calculate the number of elbows

that it will be necessary to use on the job, especially in a good sized house. You don't know just what structural difficulties you are going to encounter; you have other workmen's installations to get around and if we don't have liberal factors of safety the steam fitter will slip in a couple of elbows where they are not looked for and trouble results.

A. W. MOULDER: In the absence of Mr. Timmis, chairman of the sub-committee on hot water heating, I will answer for the Committee. To give some safe data or some comparatively simple method of figuring hot water heating pipe sizes has been brought to the attention of the Committee and we appreciate the need. Mr. Hart's statement of the trouble he has gotten into in one way or another verifies the experience of probably every member of the committee, which is composed of men interested chiefly in the technical side of the question and Mr. Timmis, myself and Mr. Offner, who have had considerable experience on the practical side.

Our decision was to draw this code very closely to the data which we had and which has been demonstrated through practical experience to give good results when carefully applied.

It seems to me that Mr. Giesecke's tests and the charts that he has shown here tonight prove at least for this one size that his tests on one inch pipe are very close to the tests made by Brabbee.

As for the practical side of it, a number of the members of the Committee feel as a result of their own practical experience that we are perfectly safe in accepting almost any one of the several authorities which we have, provided we specify in the code that the system must be carefully equalized according to that data.

We can't tell anybody how to make it easy to keep track of the number of elbows. That is the real trouble. It is, unfortunately, and I believe always will be a very difficult job to equalize a hot water heating system.

H. M. HART: I just wanted to raise a little point of caution in the work that this Code Committee does. I know a way to get around it and that is to make the pipes big enough so this question of friction of an extra elbow doesn't make so much difference.

No. 712

HEATING AND VENTILATION OF A MODERN STEAM POWER STATION

By DAVIS S. BOYDEN (MEMBER) AND A. B. WILLIAMS (NON-MEMBER)

BOSTON, MASS.

THE engineer does not as a rule associate heating and ventilation with a steam power station. The modern design of such a structure, however, does present problems of this character. These require careful study and at times are so unusual as to be almost unique. The formulation of the requirements at the new Weymouth Power Station of the Edison Electric Illuminating Co. of Boston, and a study of the methods used to fulfill them, forms the subject of this paper.

The Weymouth Power Station

The Weymouth Station has been built to carry the increasing load requirements of the Boston Edison system. It is a steam driven station of the most modern design intended for an ultimate development of from 300,000 to 400,000 kw. The initial installation has a capacity of 67,000 kw., consisting of two 32,000 kw. main turbo-generators and one 3000 kw. ultra-high pressure turbo-generator. The boiler equipment at present consists of three 2000 hp. boilers for 350 lb. steam pressure and one 1600 hp. boiler for 1200 lb. steam pressure. In common with other modern steam power stations the auxiliaries are almost exclusively of the

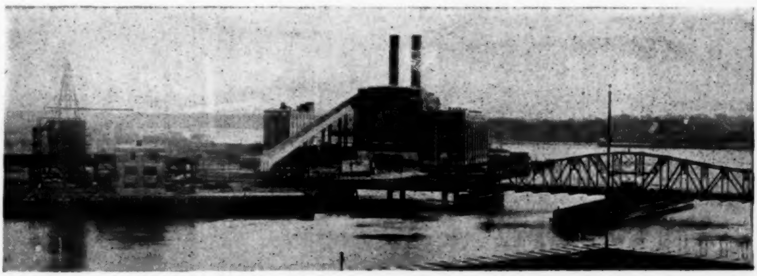


FIG. 1. WEYMOUTH POWER STATION, BOSTON, MASS.

Presented at the Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Boston, January, 1925.

electrically driven type. Such steam auxiliaries as are installed are intended for emergency use, and would be called upon in cases similar to starting the plant up from "cold" without outside power being available. One of these units is used for heating purposes as will be described more fully later. The power station buildings consist of a boiler house and turbine room built side by side with a common separating wall, and an electrical switching and control house separated some 200 ft. from the main buildings. In addition to these there are a number of small auxiliary buildings located on the station grounds. The boiler house is a brick structure 114 ft. wide and 162 ft. long in ground plan and approximately 124 ft. high. At the ground level there is a basement floor on which are located the ash handling equipment and the forced draft room. The main floor is about 30 ft. above the ground floor and forms the firing and control floor for the boilers. There are no solid floors above the operating floor, the upper floors taking the form of open platforms for the most part made of subway gratings.

The turbine room is a brick building 67 ft. wide, 196 ft. long and 90 ft. high. There is an *auxiliary* bay 30 ft. wide between the boiler house and the turbine room but the framing of the turbine room is continuous through this bay into the boiler house. The roof of the *auxiliary* bay is considerably lower than that of the turbine room so as to allow direct lighting and ventilation of the boiler house below the bunkers and on the turbine room side.

General Conditions

In a modern power station of this type, the heat available for heating the building by radiation from the power equipment is comparatively small in quantity.

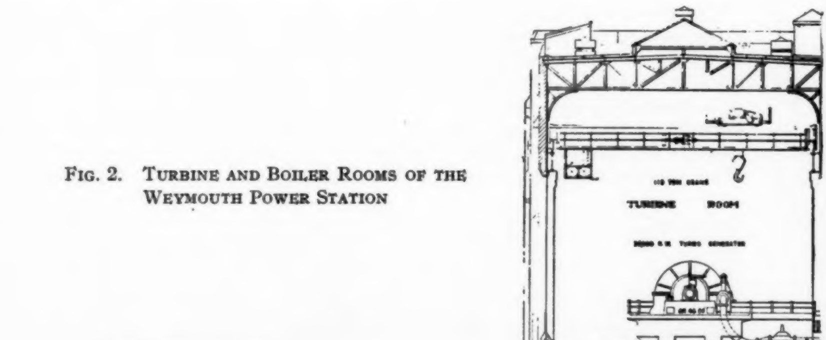
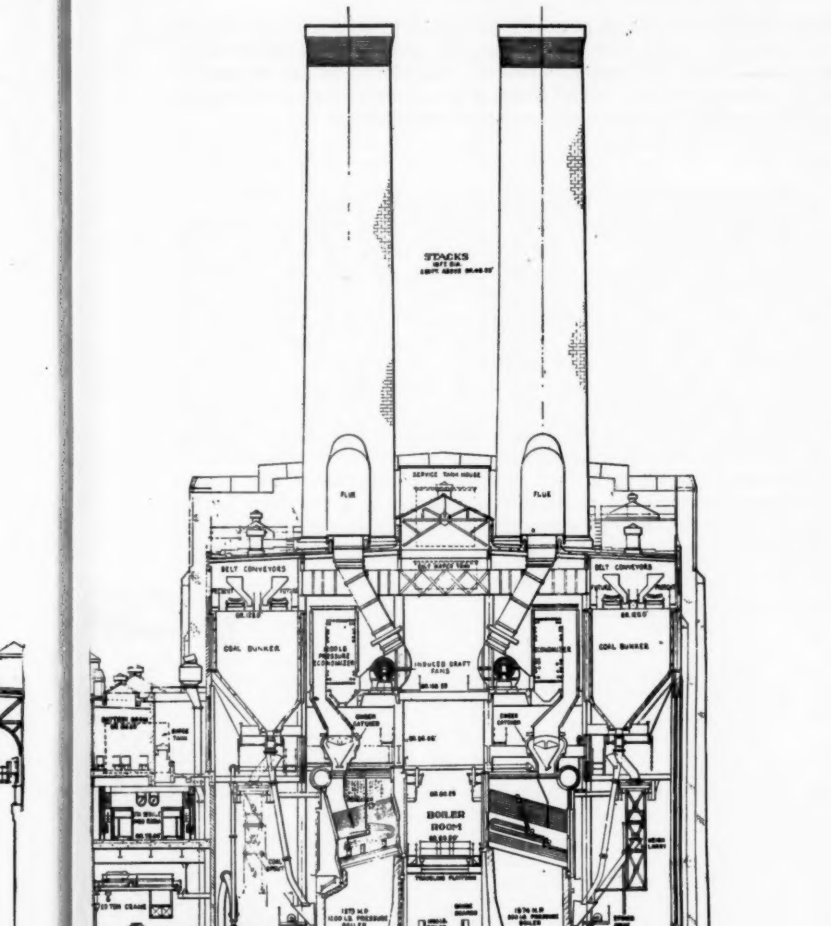


FIG. 2. TURBINE AND BOILER ROOMS OF THE WEYMOUTH POWER STATION


STACKS
FLUE
FLUE
SERVICE TANK HOUSE
BELT CONVEYORS
BELT CONVEYORS
COAL BUNKER
COAL BUNKER
INDUCED DRAFT FANS
BOILER ROOM
ASH HOPPER
ASH HOPPER
ASH ROOM
HIGH PRESSURE TURBINE
PIPE GALLERY
BOILER FEED PUMP

It is necessary, therefore, to supplement this heat by artificial heating during the severe winter weather. In summer it is necessary to provide ventilation to maintain the plant at a comfortable point. However, the heat to be carried away is so moderate in quantity that by careful design it is possible to secure satisfactory ventilation by natural circulation except in certain special cases.

Steam Supply

The heating system is based on steam supply at 10 lb. pressure. The entire plant including outbuildings is heated from a central point with a distributing system of supply and return lines. Before deciding on the source of steam supply a number of plans were considered. The one chosen is the utilization of the exhaust from a small turbine driven direct-current generator. This unit is largely supplementary to other equipment and can be operated at any load desired so that steam exhausted from its driving turbine can be made to exactly fulfill the heating requirements of the plant. To avoid a special attendant for the operation of this unit it was decided to adopt the novel plan of placing the control of the supply to the heating system and of the turbo-generator on the main switchboard. This was accomplished by installing an electrical remote indicating gage showing the pressure on the heating supply receiver with its remote indicator located on the main switchboard. Near this remote indicator is the control for the turbo-generator supplying the necessary steam. It is necessary for the operator merely to observe the steam pressure and keep it to the predetermined limits by adjusting the load on the turbo-generator. When the operator notes a

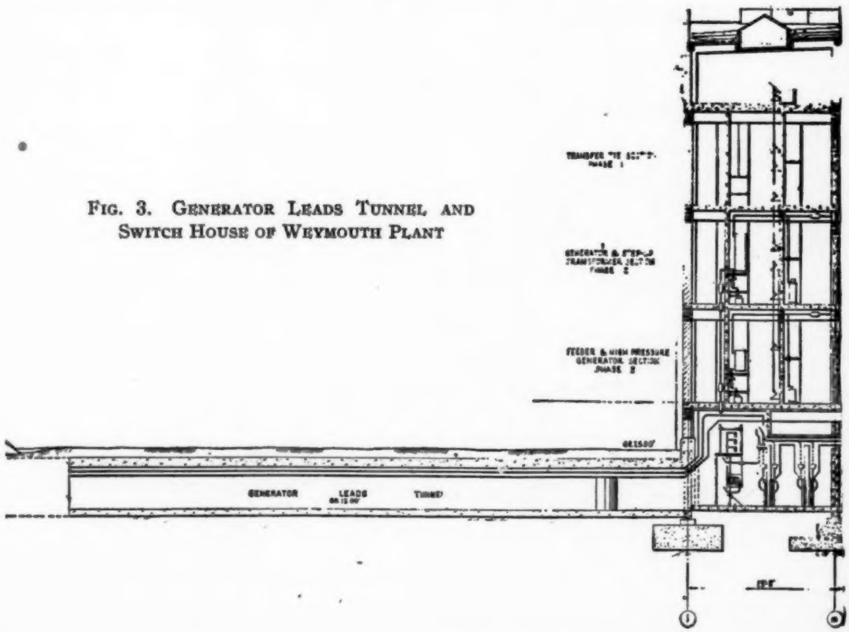
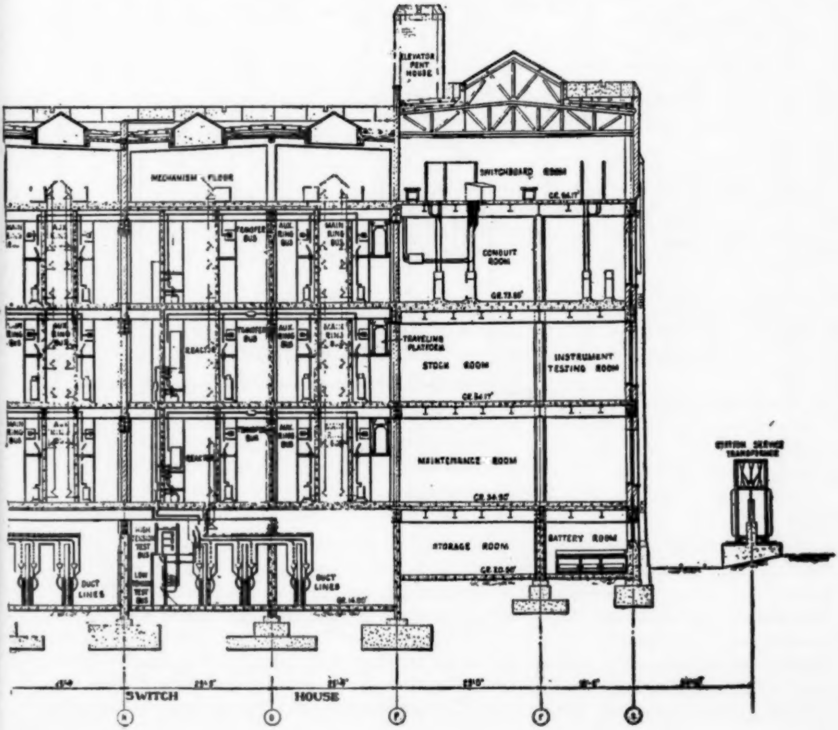


FIG. 3. GENERATOR LEADS TUNNEL AND SWITCH HOUSE OF WEYMOUTH PLANT

decrease in steam pressure he offsets it by an increase in load giving more exhaust steam and *vice versa*.

The output of this turbo-generator is utilized for station auxiliary purposes and has sufficient value to pay a considerable yearly premium. In considering this installation it should be borne in mind that practically no attendance is required nor any special apparatus as the turbo-generator used forms part of the regular station equipment.

In order to supplement the steam supply just described a special reducing valve operating on steam at full boiler pressure is installed discharging directly into the heating system receiver. The initial heating requirements indicate a maximum demand for 10,000 lb. of steam per hour. To supply this quantity of steam reduced from 350 lb. pressure to 10 lb. pressure and properly desuperheated together with the necessary nicety of regulation required a reducing valve of unusually excellent characteristics. The valve used consists of a 3 in. extra heavy steel body angle valve. The seat has been replaced by a carefully designed Monel nozzle of the stream line type. The disc of the valve has been replaced by a combination needle valve and disc. This needle valve in combination with the nozzle provides steam throttling passages of stream line contour, at practically all capacities from zero to maximum flow. A further refinement is that the effective throat of the



nozzle is maintained well away from the disc seat so that the valve can operate under severe conditions without erosion due to "wire drawing." The valve is operated by a motor and reduction gear. This provides an extremely powerful positive motion and effectively prevents any possibility of chattering. The motor is controlled by a reversing relay panel operated by a pressure gage connected to the receiver tank. The gage and control are adjusted so that the valve begins to open when the pressure in the tank drops to 7 lb. If the pressure continues

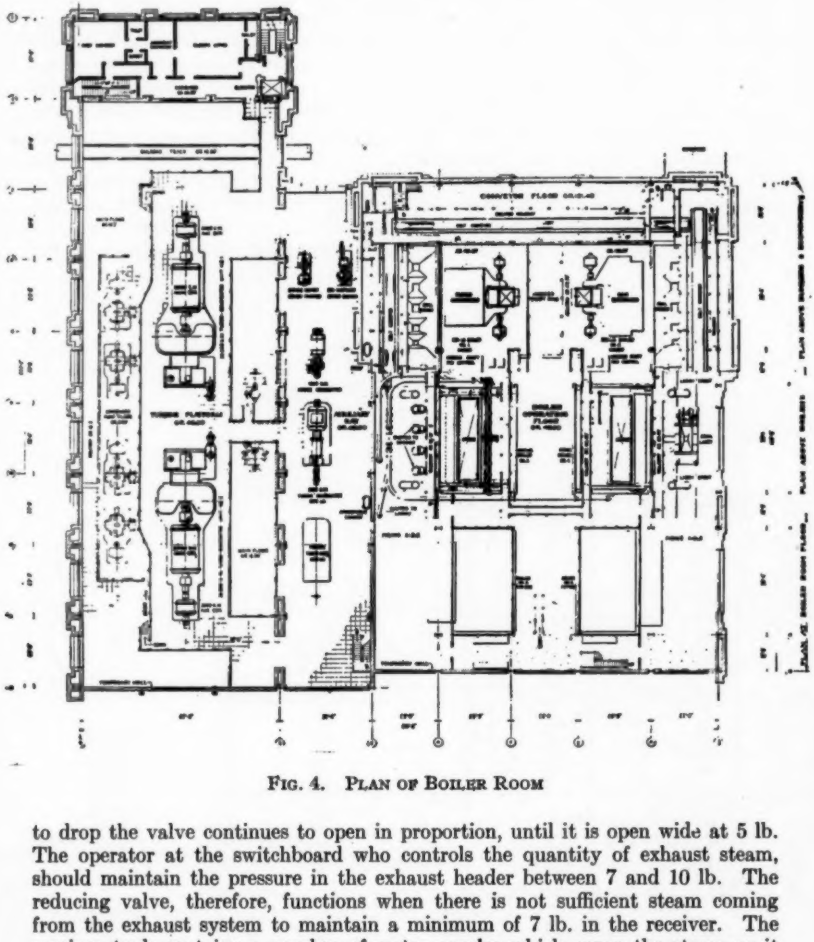


Fig. 4. Plan of Boiler Room

to drop the valve continues to open in proportion, until it is open wide at 5 lb. The operator at the switchboard who controls the quantity of exhaust steam, should maintain the pressure in the exhaust header between 7 and 10 lb. The reducing valve, therefore, functions when there is not sufficient steam coming from the exhaust system to maintain a minimum of 7 lb. in the receiver. The receiver tank contains a number of water nozzles which spray the steam as it passes through the tank, thus providing sufficient evaporating effect to destroy the high superheat which would otherwise be present. The low pressure system

is protected from possible excessive pressure by suitable safety or back pressure valves.

Return System

The heating system is laid out on the two-pipe principle, with thermostatic traps on the return end of all radiating fixtures. The condensate is conserved in all cases and returned, in general, by gravity to a concrete heating return tank located alongside the distilled water tank in the power station. There are no vacuum return pumps. The water in the heating return tank is returned to the

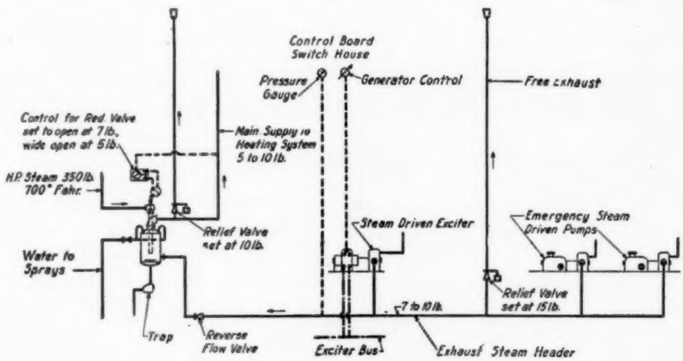


FIG. 5. STEAM SUPPLY FOR HEATING SYSTEM

main power station feed system by a horizontal motor driven centrifugal pump operating under float control. Should this pump be out of service for any reason, the water overflows to the station distilled water tank where it is returned to the main feed system by the distilled water pumps, in accordance with make-up requirements of the boilers.

The system is simple, requires very little operating attendance, and conserves the returns along with the condensate from the other power systems. This last is desirable as all make-up is distilled water supplied from evaporators.

Turbine Room

In the turbine room the heat generated by the power equipment is in general sufficient to maintain a satisfactory temperature. The large areas of glass in the full height windows require special treatment as they would otherwise produce undesirable cold-air currents. To counteract the loss of heat from these windows direct radiation is installed in wall recess under the windows. The warm air from this radiation is discharged through grilles immediately below the glass. Sufficient radiation is provided to completely temper the air in most severe weather.

A problem characteristic of power station work is the condensation of moisture on the underside of the roof. At the Weymouth Power Station this has been cared for by placing insulating material in the roof slab so as to prevent heat loss and chilling of the underside of the roof with the consequential condensing of the water bearing vapors. This has been supplemented by the careful prevention

of steam leakage and vapor production which would form a source of excessive humidity in the turbine room area.

At the entrance of the power station the turbine room is extended to form an office building. Here are located the executive offices and rooms for clerical work together with laboratories, and in the future dining rooms and sleeping quarters. The heating system in the office building follows the usual design for this class of work. The heating is of the direct radiation type, largely single column supported by brackets on the walls and without legs. No attempt is made for special ventilation other than the natural ventilation found in a building of this class.

Boiler Room

In the boiler house the windows have been treated with direct radiation as in the turbine house except that the radiation here is not concealed. The ash basement

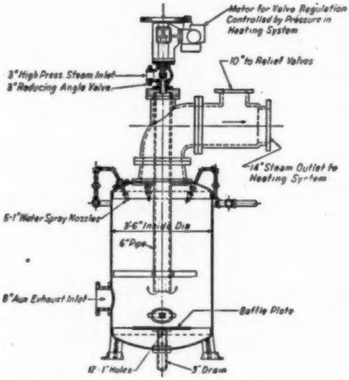


FIG. 6. REDUCING VALVE—DESUPERHEATING TANK

in some plants is a source of extreme cold if the forced draft fans draw their air from this space. At Weymouth, the fans are located in a single long fan room partitioned off from the rest of the basement. The air inlets to this room are through louvres in the outside wall. In this way the low temperature and high velocity air currents are confined. In the future it is proposed to install ventilating fans near the top of the boiler house which will draw the warm air from this point and force it through ducts into the forced draft fan room. In this way the tendency for warm air to collect in the upper part of the boiler room will be eliminated, and the fan room will be tempered and the warm air made available for combustion purposes. In the boiler house the absence of all floors above the operating level allows excellent air circulation. The boilers are arranged in two rows with a central operating aisle between. The stokers are located in the outer aisles. In this way direct light and ventilation is provided for the stokers and coal handling equipment. Furthermore these aisles are separated from the remainder of the boiler house by partitions which confine the coal dust to that portion of the plant. The operating aisle is kept clear of galleries and is opened directly to the

skylight in the roof over the boiler house. Louvres in the side of this skylight provides the necessary ventilation. This is augmented by swinging sash in the boiler house.

Smoke uptakes and flues together with cinder catchers, economizers, and induced draft fans are carefully insulated so that radiation is reduced to a minimum. For this reason the temperatures in the upper part of the boiler house will probably not be uncomfortably high at any time. Provision is made, however, for the future installation of exhaust fans which will discharge large quantities of warm air from the economizer floor level to the forced draft fan room as already mentioned.

Auxiliary Bay

The auxiliary bay has certain spaces such as battery room and auxiliary switch room in which direct radiation is not desirable for electrical safety reasons. These spaces, however, require both heating and ventilation. To accomplish this, a supply fan with a bank of vento heaters and a system of supply and exhaust ducts have been installed. In this way the temperatures can be maintained and ventilation provided in both summer and winter. The obnoxious fumes from sulphuric acid in the batteries make the ventilation requirements of the battery room severe, and a special exhaust fan has therefore been installed. The fan and ducts for the battery room service are lead covered.

Aside from the space in the auxiliary bay already mentioned, ventilation is arranged through natural air flow. The main turbine room is ventilated by louvres placed in the turbine house roof and air is supplied through swinging sash in the main windows. For convenient operation the sash are controlled electrically through motor operating gear.

The auxiliaries in normal operation in the turbine room are motor driven, and therefore, give off much less heat than would the corresponding interlocking steam driven equipment. Furthermore, these auxiliaries are grouped in open galleries so that a free circulation of air is provided at all times.

Switch and Control House

The switch and control house forms a separate problem. This building houses the high voltage electrical switch gear and contains the main control room. None of this equipment must be endangered by having piping in its proximity, or the possibility of steam and water leaks. In aisles or other spaces not containing high voltage equipment, direct radiation is installed and ventilation supplied by swinging sash in the windows. The high voltage equipment is located on three floors and arranged in longitudinal aisles. These floors must be kept isolated from each other. Certain portions of this equipment generate heat which must be removed in accordance with maximum allowable temperatures. The entire building must be maintained at a temperature suitable for reasonable working conditions, and also to prevent any possibility of moisture being condensed on the electrical structures and equipment. The operating room must have a comfortable temperature at all times.

In general this building is filled with a mass of electrical equipment, buses and cable, the arrangement and location of which control the general space requirements. The heating and ventilating system has to be subordinate to these but by attacking the problem at an early period in the plant design and on a broad comprehensive basis, it is believed that a system has been installed which accomplishes the purpose and yet is very simple.

Switch Bay

The steam for heating is drawn from the central source in the power station. Supply and return lines run underground and are placed in insulated tile conduit with suitable drains, anchors, expansion joints, and manholes.

The switch bay containing the phase rooms and mechanism floor is heated and

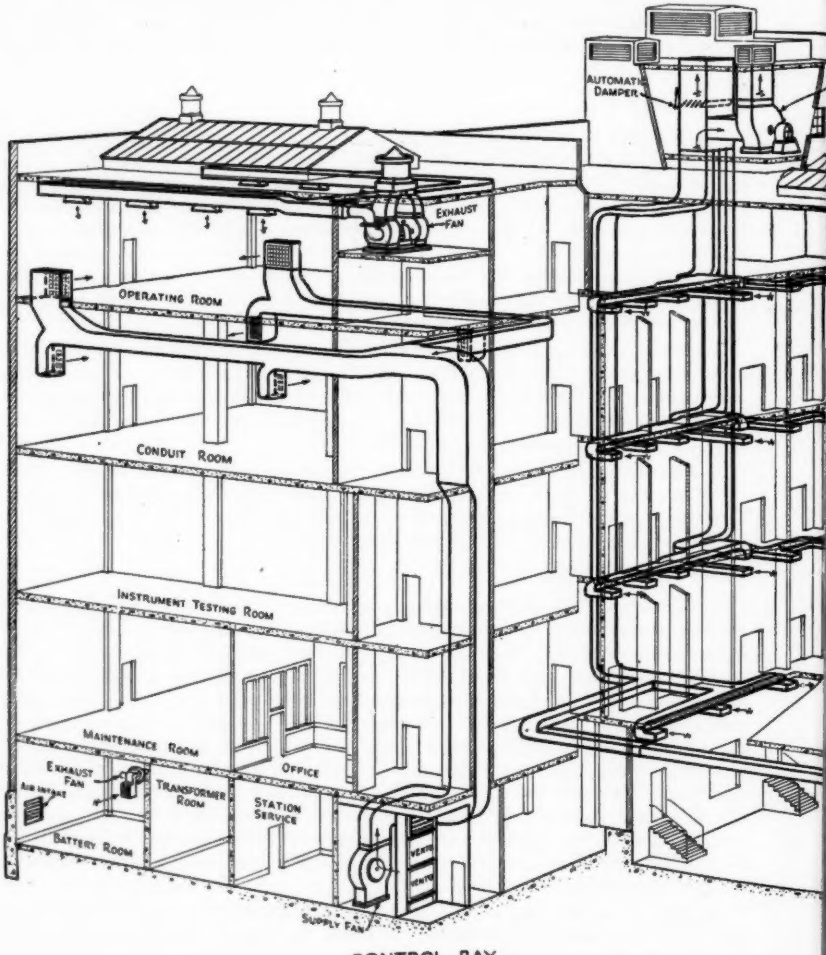
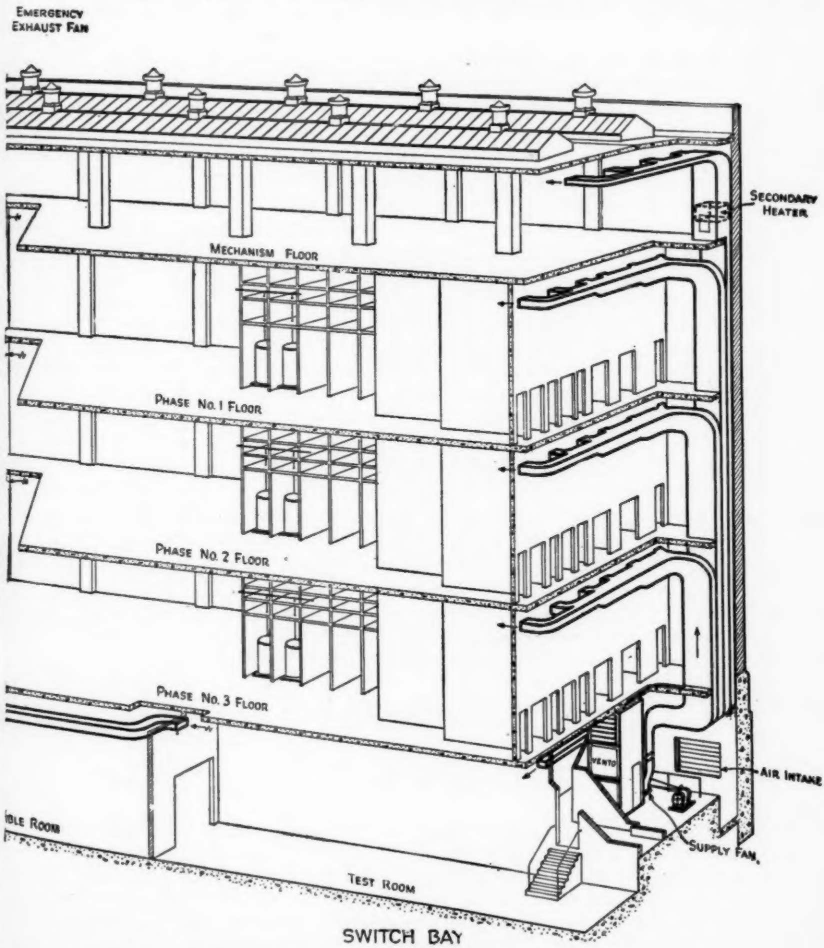


FIG. 7. LAYOUT OF VENTILATING DUCTS IN CONTOL BAY A SWI

ventilated by a system of vertical supply and exhaust ducts. These carry air which performs both functions; that is, heating in winter and "flushing out" or cooling in summer. There is a supply fan and vento heater at the base of the supply system and an exhaust fan at the top of the exhaust system. Each floor has its own vertical supply and exhaust ducts with branches run to one end of the phase aisles. The supply ducts admit air to one end of these aisles and the exhaust ducts remove it at the other end. No considerable longitudinal duct system has been installed. The aisles themselves are utilized to pass the air from the vertical supply system to the corresponding exhaust system. Ducts in the aisles



Switch Bay of the Weymouth Power Station, Boston, Mass.

TABLE 1. HEATING AND VENTILATION—SWITCH HOUSE—SWITCH BAY, WEYMOUTH POWER STATION, THE EDISON ELECTRIC ILLUMINATING CO. OF BOSTON

Supply Fan Capacity 35,000 c.f.m., $1\frac{1}{4}''$ Water, 25 hp. belted motor, local control.
Primary Vento Heater 1123.5 sq. ft., heating surface, 0–81° F.
Secondary Vento Heater 193.5 sq. ft. heating surface, 70–114° F.
Emergency Exhaust Fan Capacity 48,000 c.f.m., $1\frac{1}{4}''$ Water, 35 hp., direct connected motor, remote control.
Min. winter temp. 0° F. Max. summer temp. 100° F.

		Winter conditions						Summer conditions			Air requirements		
		A		B	C	D	E	F	G	H	I	J	K
Floor	Room	Allowable Inside Temp. Range Min.	Max.	Heat Generated B.t.u. Hr.	Heat Loss B.t.u. Hr.	Make-up Required B.t.u. Hr.	Air Supply Temp.	Heat Generated B.t.u. Hr.	Heat Loss B.t.u. Hr.	Heat to be Removed B.t.u. Hr.	*Climatic Conditions Determining Air Supply	Air Supply Gravity Exhaust C.F.M.	Emergency Fan Exhaust C.F.M.
	Test	50°	100°	None	26,200	26,200	80°	None	None	Negligible	Winter	800	1,810
Basement	Cable	50°	100°	None	16,350	16,350	80°	None	None	Negligible	Winter	500	1,270
	Cable	50°	100°	None	16,350	16,350	80°	None	None	Negligible	Winter	500	1,270
First	Aisle 1	50°	115°	6,830	31,300	24,470	75°	6,830	6,830	0	Winter	900	2,030
	Aisle 2	50°	115°	6,830	Negligible	None	75°	6,830	Negligible	6,830	Summer	420	1,160
	Aisle 3	50°	115°	6,830	Negligible	None	75°	6,830	Negligible	6,830	Summer	420	1,160
	Aisle 4	50°	115°	13,660	Negligible	None	75°	13,660	Negligible	13,660	Summer	840	1,900
	Aisle 5	50°	115°	46,600	Negligible	None	75°	46,600	Negligible	46,600	Summer	2,850	5,550
	Aisle 6	50°	115°	3,415	39,400	35,985	75°	3,415	3,415	0	Winter	1,320	2,750
Second	Aisle 1 to 6	50°	115°	Summer and winter heat conditions similar to first floor								6,750	14,550
Third	Aisle 1 to 6	50°	115°	Summer and winter heat conditions similar to first floor								6,750	14,550
Fourth	Mechanism	50°	100°	None	294,700	294,700	112°	None	None	Negligible	Winter	4,340	None

* The volume of air for heating or "flushing out" is determined by which ever climatic condition requires the greater amount. This same amount of air being supplied at all times therefore takes care of the other climatic condition.

The volume of air when the emergency fan is in operation is in excess of these requirements in order to provide for unusual heat or smoke conditions.

themselves would have been objectionable due to proximity to electrical equipment and would also have been difficult if not impossible to install.

Flushing Out Principle Used

The complexity in the requirements of heat loss and heat generated has been handled in a very simple manner by utilizing the "flushing out" principle. In this, all compartments, some of which generate heat and some requiring heat, are brought to the same general condition by flowing through them, air of a moderate temperature; that is approximately 75 deg. in winter, or the outdoor temperature in summer.

The exhaust system normally carries away the heat by natural gravity ventilation induced by the difference in outside and inside temperature. This gives simplicity of operation, and under normal operation eliminates equipment requiring supervision and maintenance.

For possible extreme heat conditions this exhaust system is augmented by the exhaust fan on the roof which can be used to increase the amount of heat and air removal. In addition this fan will be used to hasten the removal of smoke in event of a fire or short circuit.

This fan is located in a penthouse on the roof with its suction connected to the main ventilating shaft, and its discharge to the atmosphere. It is arranged so that when the fan is started a damper in the gravity ventilating shaft is shut automatically, and in doing so the air is by-passed through the fan to the atmosphere instead of going directly outdoors as in the case of natural ventilation. For emergency service, especially the removal of smoke, the fan is started and the damper closed from push button stations on each floor. These push buttons are interlocked so that the equipment can be stopped only from the station where it was started.

It is not expected that the distribution of air when the exhaust fan system is in operation will be in exactly the same ratio as with gravity exhaust. The amount of air from all parts of the system will be increased in a relation accurate enough for practical purposes.

No dampers except the automatic one at the exhaust fan are manipulated at any time. They are set for proper distribution and then locked in position. For electrical operating reasons the doors to the different electrical compartments are normally closed and locked. The ventilating system, therefore, remains in balance and at only one place requires operating attention; that is, the supply fan and heater.

A tabulation is attached showing the heat lost, generated, and removed from the switch bay.

Control Bay

The control bay operation of the switch house is treated in somewhat the same general manner except that only a small portion contains "live" electrical equipment that requires indirect heating and ventilating. The major portion is, therefore, heated with direct radiation. The battery room is a local proposition and is treated as such by having independent openings for air and exhaust with an auxiliary fan for use when there is a large amount of dangerous gases during periods of battery charging.

Breaker House

The breaker house, which is a small brick building containing coal breaker, office, and toilets for the coal handling force is heated by direct radiation, ventilation being of the usual natural flow through hinged sash.

The steam supply is secured from the central source in the power station, with the supply and return lines carried in the structure which houses the inclined coal conveyor. By locating the lines in this housing in preference to underground, a considerable saving was secured in installation cost. Also less heat is lost, or at least the heat is put to some advantage by moderating the temperature in the gallery. A motor driven float controlled pump returns the condensate to the central return tank in the power station.

Conclusion

In general the aim has been to secure a system that is simple, and that requires the minimum of operating attention. Fans and pumps have been dispensed with wherever considered possible and gravity used to a considerable extent. The design was worked out in conjunction with the power problems and not as an afterthought. The design of the building and arrangement of equipment was utilized to solve many of the heating and ventilating problems by eliminating them.

DISCUSSION

PRESIDENT ADDAMS: We are certainly indebted to these co-authors for their interesting paper on this unique problem. The paper is now open for discussion.

W. H. CARRIER: I would like to ask the particular purpose in mind for the elaborate painstaking care shown in the design of the reducing valve. It would be enlightening to know the reasons for it and how it works out.

A. B. WILLIAMS: I will call on Mr. Norris of our organization to answer that. He handled all the details of the work at that time.

E. W. NORRIS: The reducing valve was developed particularly for this work as it was felt that the requirements of a large pressure drop from 350 lb. per sq. in., where there was a total temperature of 700 deg. Fahr. was beyond the capacity of the ordinary type of valve. In developing the present valve, great care was exercized in avoiding the possibility of wear on the valve which would have to operate through a long period of service. Use was made of the development already at hand in steam turbine work and the nozzle design and needle control already demonstrated for steam turbines was used. The motor operating gear was put on to provide an absolutely definite control which would be perfectly free from chattering at all loads when the valve was in service, and so far it has been perfectly free from chattering. Of course, it is too soon to say how the wearing qualities are going to be, but its operating qualities are fully demonstrated.

S. A. JELLETT: I noticed in the description that the minimum pressure carried is 7 to 10 lb. You are carrying exhaust steam in that receiver at 7 lb., as I take it, and then adding the live steam through this reducing valve. It is my own experience with reducing valves that we have never yet successfully stepped down from 300 to 5 or 7 lbs. with one valve. We have to break it two or three times and expand it on the pressure side of those valves.

I am interested to know just what the result will be here under those conditions—

injecting live steam at 7 lb. pressure into a receiving chamber in which there is exhaust steam at 7 lb. and out of which the flow is not constant because the heating conditions vary. I would like to have brought out the question as to just why this receiver and valve were put in combination this way and what the effect of increase or decrease in condensation of the heating system is.

E. W. Norris: The point brought out is very well taken and, as I briefly indicated before, in the design of a special valve of that type it has not previously been possible to drop the whole way with one valve, but by utilizing a stream line nozzle to control the steam from a high initial pressure to any final pressure up to a perfect vacuum it is entirely possible to do so without wear and tear on the nozzle.

The pressure flow through the nozzle is interesting. The initial pressure is 350 lb. At the throat of the nozzle the pressure is about 58 per cent of the initial. That is characteristic of all nozzles. From the throat to the final exit the pressure drops from 58 per cent to the exhaust pressure, whatever that happens to be. In this case that is determined by the pressure on the system. The receiver acts as a storage tank between the exhaust system and the heating system at the plant. Steam flowing has an opportunity to steady the operation of the control mechanism.

The action of the valve is not affected by the flow of steam because it is not a balanced valve and by using motor operating gear the forces controlling the position of the valve are so much larger than the steam forces that the action of the valve is perfectly smooth in any position.

A. L. Becker: I would like to ask if the sprays in the receiver are controlled by this same reducing valve so they are wholly in operation when the high pressure steam is being introduced.

A. B. Williams: The sprays are not controlled automatically in any way. They are put on by hand because the exhaust steam also contains a very high amount of moisture and the difference in temperature between the exhaust steam and the high pressure steam was not great enough to make us want to bother with more fixtures that might get out of order. That is another case of simplicity of operation; the sprays are on when the heating system is in operaton and left on. The excess water in the receiver, if there is any, is drained off automatically and put back into the distilled water system, thus conserving both the water and its heat.

W. H. Carrier: A suggestion might be made, if it is not out of order, that it is quite possible to get a very good saturation on all reducing valve loads, by the introduction of a spray from a maintained water level. I have had occasion to accomplish the same purpose without any noise or other trouble. It makes it entirely automatic and avoids the use of sprays. I would think that for power plant work, such a system would be very excellent.

I might ask if anything of that kind has been considered.

A. B. Williams: It is quite a novel suggestion. I certainly would like to know more about it. In the pressure of designing the power station, we keep to the simplest things as far as possible. I have no doubt but that there are certain parts of that heating system that can be improved and we certainly hope to receive suggestions.

President Addams: Is Mr. Seabury, the superintendent of the Engineering Department of the Edison Electric Company in the room? May we hear from you?

G. E. SEABURY: Mr. President, I am here in a dual capacity. Your secretary very courteously extended an invitation to the *National Electric Light Association*, through its manager in New York, Mr. Aylesworth, to be present at your meeting and that invitation was passed down to me.

This is a most interesting paper and it has been very well handled. There are two reactions which occur to me in connection with it.

The heating and ventilating installation certainly indicates the progress made in power plant design as well as in heating and ventilating. It is not so long ago that no attention was paid whatever to ventilating or heating of power houses. A few years ago I had something to do with a power station just outside of Paris. The whole of one side of the station was built in the form of a louvre and in extremely cold weather they had bonfires built on the floor of the power house to make it habitable. We have recognized the problem of heating and ventilating as a problem connected with the design of power plants.

Another feature brought out and not usually considered in problems of ventilation is that we may have oil fires or fires produced by short circuits, and in order to make repairs quickly it is necessary to get the smoke out of the room so that men can get in there to work.

I wish to extend to the members of this gathering an invitation to visit the power plant at Weymouth.

PRESIDENT ADDAMS: That is certainly a fortunate invitation. If all is true that has been said here and that my old friend Mr. Fred Lowe observed when he was in Boston a few weeks ago and told me about while going down to New York, there is something in store for us, I assure you.

No. 713

THE INDUSTRIAL APPLICATIONS OF OZONE

By FRANK E. HARTMAN[1], SCOTTDALE, PA.

MEMBER

THE value of an oxidizing agent that produces no contaminating residue is so thoroughly realized, that ozone investigations excite unfailing interest. Ozone, for many years, has held forth possibilities that have been realized only in part. Unfortunately, there has been much misleading information published about ozone, and a vast majority of its literature is either purely qualitative or emanates from academic investigations conducted along lines that hold but little interest for the practical engineer. A simple treatise on the practical applications of ozone does not exist in a single volume; and its Journal literature is badly scattered and highly specialized. A quantitative, analytical treatment of the subject, written from the view point of the engineer, is badly needed, and while it is without the scope of a paper such as this to fill the want, some indication of the present, practical status of ozone will be attempted.

The chemical properties of ozone are all that can be desired of an oxidizing agent. They are too well known to spare them space here, but rather shall an attempt be made to analyze its more practical applications. The efficiency of ozone production has always been known to be very low, however, about even so fundamental a piece of datum as efficiency of production, there seems to be a great deal of question where ozone is concerned.

If the production of ozone is considered as an electrolytic reaction, the theoretical yield should be 0.0005 gram per coulomb, whereas from 0.2 to 0.3 gram per coulomb have been obtained. Obviously the electrolytic theory must be dismissed from consideration; since efficiencies above 100 per cent have not, so far, been visited upon the efforts of man. The formation of ozone is a strongly endothermic reaction. If calculation of the electrical energy supplied to an ozonizer to heat is made, and the heat of formation of ozone as 34,000 calories per gram mol is considered, it is found that best results yield the surprisingly low efficiency of 5 per cent when air is used as the source of oxygen, and 15 per cent when pure oxygen is used. This indicates that with air, 95 per cent of the energy supplied an ozonizer is dissipated as heat. Using the apparatus shown in Fig. 1[2], one can readily show this to be the case. With this apparatus, the electrodes consist of flow streams of dilute acid. Thermometers are so arranged as to indicate the temperature of the inflowing and out-

[1] Chief Chemist, United States Ozone Co.
[2] Designed by Dr. Alfred Starke, U. S. Ozone Company, Research Staff.

Presented at the October, 1924, meeting of the New York Chapter, AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS.

flowing acid, together with the inflowing and outflowing air, an insulating jacket (not shown) is, also provided. The quantity of air is measured volumetrically, the acid is carefully weighed and the quantity of ozone produced determined chemically. From these data the energy used in the formation of ozone can be calculated, together with the energy absorbed by the acid cooling media and the air. The sum of these energies checks, within the range of observational error, with the energy input, after deducting the transformer losses, as measured electrically. This affords an excellent means of determining the actual efficiency of any ozonizer, and the gross efficiency, which includes the losses in the transformer, etc., of a good, ozonizer producing ozone in concentrations of from 5 to 6 grams per cubic meter, is about 5 per cent or approximately 60 grams per kilowatt hour. A pound of ozone will then require about 7.5 kilowatt hours of energy. At the figure generally asked for electricity, ozone can be produced at from 20 to 30 cents per lb.; figuring gross efficiencies and interest on investment. Now an oxidizing agent of the value of ozone can find numerous applications at a cost even greater than 30 cents per lb., yet the application of ozone has hitherto been somewhat limited. Comparatively speaking, the cost of ozone apparatus may be considered as being high. It has previously been stated that a good yield is in the neighborhood of 60 grams per kilowatt hour. Now an ozone generator is limited as to size, since the material requirements are very exacting homogeneous dielectrics, above certain dimensions, cannot be obtained. The largest generator that it is practical to build will, when operating on 60 cycle current, have an energy absorption of from 85 to 150 watt hours. That means that it will yield from 5 to 9 grams of ozone per hour. The total anode surface of this generator is approximately 250 sq. in., hence its *energy density* is a matter of from 0.34 to 0.60 watt hour per square inch. It is this very low energy absorption of ozonizers that contributes to their cost. Adequate cooling can be accomplished economically, by artificial means, to handle *energy densities* of 30 to 40 times these quantities, hence the advisability of increasing the energy absorption of ozonizers where large quantities of ozone are required, can readily be seen.

An elevation of the potential difference between the electrodes will increase the energy absorption of an ozonizer, however, since it is necessary to employ a solid dielectric, there is a decided limit to the potential difference, as it is not good practice to approach within 50 per cent of the perforation tension of the dielectric material used. An elevation of the temperature of the gas will increase the energy absorption, but at elevated temperatures thermal decomposition of ozone is favored, hence there is a decided limit to this factor, in fact better yields are obtained with cold air. If one would increase the energy absorption of an ozonizer one must first consider the electrical characteristics of an ozonizer circuit. Fig. 2 illustrates a typical ozonizer circuit. The secondary consists of a capacity in series with an induced source of power, hence the capacity is much greater than the inductance and the power factor will lead by a considerable angle, unless the impedence of the transformer is especially high. Practically the entire resistance of such a circuit is a capacitive reactance. Capacitive reactances are proportionate to:

$$X_c = \frac{1}{2\pi f_c}$$

where X_c = the capacitive reactance in ohms; 2π = 6.283 and F = the frequency, or $2\pi f$ = the angular velocity in radians per seconds; C = capacity in farads.

Experiment has established that capacity is little affected by even exceedingly wide variations in frequency. From the equation it follows that the resistance of

ozone circuits is inversely proportionate to the frequence. From Ohms law the current is inversely proportionate to the resistance. The sum of these facts then reveals that the energy absorption of an ozonizer is directly proportionate to the frequency.

A search of the literature revealed that many workers had experimented with high frequencies, and many patents had been issued covering high frequency ozonizers.

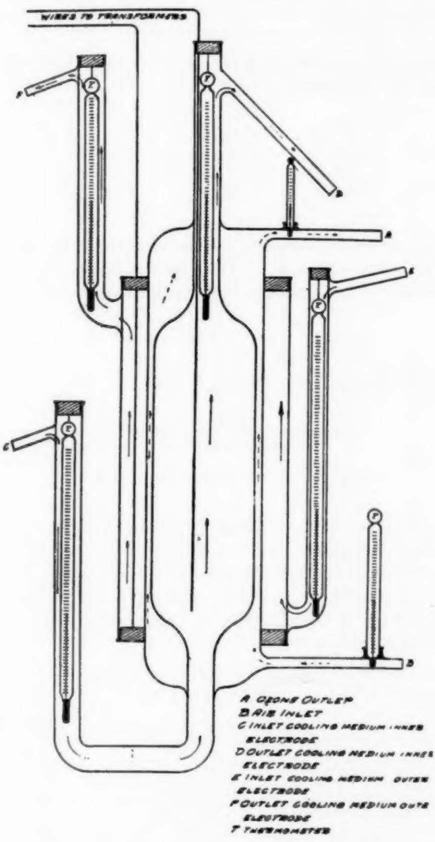


FIG. 1. APPARATUS TO DEMONSTRATE DISSIPATION OF HEAT IN OZONE PRODUCTION

The early workers had, however, employed damped waves, since at that time sustained waves of high frequencies were not available. The small available energy from damped waves renders them out of consideration in industrial applications. The advent of the radio-telephone brought into existence high frequency alternators that produced sustained waves. Their earliest application to the

production of ozone was made in 1914 by Puschin and Kauchtschev.[3] Their work tended to reveal a connection between the frequency employed and the applied potential. In this instance, they cite the following as essential to equivalent yields:

Frequency	Potential
1240	6500
950	7000
660	8000

An arbitrary calculation based on good ozonizer practice, reveals that the energy density would stand in the following relation:

Frequency	Potential	Watts per Tube
1240	6500	32
950	7000	29
660	8000	25

The wattage figures are in sufficiently close agreement to indicate that its influence is probably the controlling one. One must always consider the other factors which enter. Puschin and Kauchtschev's report did not contain the data for estimating the extent to which they may have influenced results. Our work indicates that potential, taken separately, is with difficulty linked with ozone yield, while energy density is strongly linked. The work of the Russian investigators further points out that for a constant air flow an increase in frequency above 1240 cycles tends to decrease the ozone yield, while an increasing air flow displaces the maximum toward increasing frequencies.

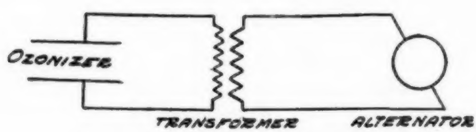


FIG. 2. TYPICAL OZONIZER CIRCUIT

Our early work[4] with high frequencies bore out this statement, however, it was found that the decrease in yield, with constant air flow, was due to thermal decomposition of the ozone, and, if adequate cooling could be effected with increasing frequencies, higher concentrations and yields could be obtained. Fig. 3 shows graphically the average results of hundreds of experiments with frequencies ranging from 60 to 1200 cycles. Fig. 4 shows graphically the average of many experiments conducted with frequencies ranging from 60 to 5000 cycles. In both of these graphs, the straight line shows the theoretical increase, as calculated by the equation given on page 712, and it is interesting to note how closely the plotted data conforms to the theoretical values.

Referring to Fig. 4 it can be seen that the increase in energy absorption of an ozonizer is a straight line function of the cycles and going from 60 to 1000 cycles the energy absorption is increased sixteen and two-thirds times. This means that an ozonizer operating on 1000 cycles current will produce $16^2/_3$ times more ozone

[3] *Journal Russ. Phys.-Chem. Soc.*, **46**, 576 (1914).
[4] Conducted in 1919—by the writer.

than the same generator operating on 60 cycles, since the production of ozone is a straight line function of the energy input. If we go from 60 cycles to 5000 cycles the ozone output will be increased 78 times. Now let us see just what this means in the terms of dollars and cents.

Assuming that an ozonizer can be built for operation on 60 cycles current for units cost, our experience reveals that an ozonizer can be built for operation on 1000 cycles for two units of cost. Then if unity ozone can be obtained for unity cost on 60 cycles, on 1000 cycles 8 units of ozone can be obtained for unity cost. An ozonizer operating on 5000 cycles can be built for three units of cost, therefore by using 5000 cycles 26 units of ozone for unity cost can be obtained. The cost here

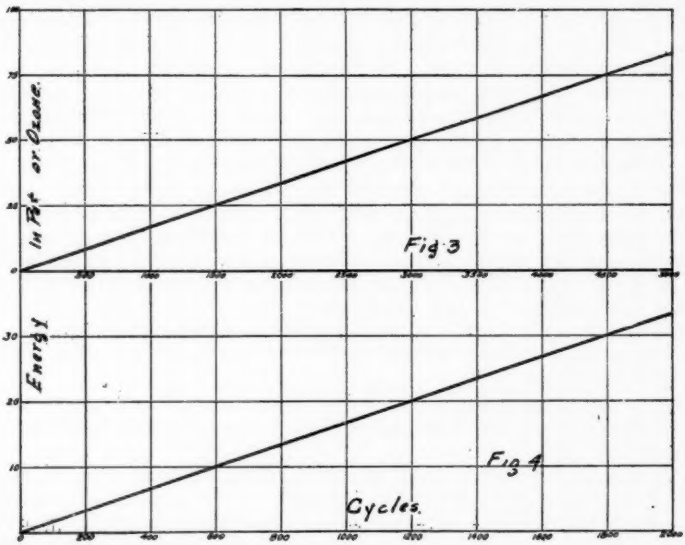


FIG. 3. GRAPHIC REPRESENTATION OF EXPERIMENTS USING FREQUENCIES OF 60 TO 1200 CYCLES

FIG. 4. GRAPHIC REPRESENTATION OF EXPERIMENTS USING 60 TO 5000 CYCLES

considered is confined to the ozonizer only. Since commercial frequencies do not exceed 60 cycles, in order to obtain higher frequencies special high frequency alternators must be provided. Assuming that unity ozonizer cost, exclusive of control apparatus, etc., is $200, then, in the case of the 1000 cycle ozonizer, $16^2/_3$ times unity cost would be $3333, but as the ozonizer will cost only $400 we show a net saving in ozonizers of $2933. Since it has been seen that the energy absorption of a single ozonizer unit at 60 cycles amounts to 150 watt hours at 1000 cycles the energy absorption will be 2.5 kw.hr.

A thousand cycle motor generator set of 2.5 kw. capacity can be purchased, with all accessories to adapt it to ozone work, for approximately $1500. From this it will be seen that even in so small a plant a saving in investment of nearly $1500 can be effected. Cycles higher than 1000 will show a much more substantial saving,

and as the price of larger capacity higher frequency alternators is proportionately smaller than the small sizes, a very substantial saving can be shown at one thousand cycles in plants of from 50 kw. and up.

At present we are placing on the market two sizes of 1000 cycle ozone equipment, a $2^1/_2$ kw. plant producing approximately 150 grams of ozone per hour, and a 5 kw. plant producing 300 grams, or approximately two-thirds of a pound, of ozone per hour. The efficiency of the high frequence end of the equipment is as good as with 60 cycle ozonizers, however, due to the losses in the motor generator set the overall efficiency of the plant is a little less than with the 60 cycle equipment. In large installations, producing their own power, this loss in efficiency can be eliminated by installing the high frequency alternator in the power house without going through a motor generator set, and as this power would have to be furnished in any event, part of the cost of the high frequency alternator can be deducted from the cost of the ozone equipment and thus a greater saving can be shown.

Due to the difficulty of obtaining standard alternators producing frequencies above 1000 cycles a 5000 cycle equipment is not ready. A great deal of work is now being done in our research laboratory on higher frequencies, however, and it is only a matter of time before higher frequencies will be available.

High frequency ozonizers are of interest only where ozone is required in quantities exceeding 150 grams per hour. Up to this capacity ordinary commercial frequencies can be utilized to a greater advantage. There are numerous applications for low frequency ozonizers, and it is doubtful if high frequency ozone will ever find application in ventilating, outside of cold storage work, since a very little ozone accomplishes a great deal of work in mechanical ventilation. Two-hundred thousand cubic feet of air per minute will require no more than 35 grams of ozone per hour, for ordinary conditions, and never more than 70 grams per hour for conditions where very pronounced odors exist.

For the sake of completeness a few of the fundamental factors of ozonizer design will be touched upon before passing on to some of the industrial applications of ozone.

A product is generally not much better than the raw material from which it is made. As ozone is produced from oxygen, and as the air is generally the source of oxygen used, there are some very exacting requirements of the air that is utilized by an ozonizer. Water vapor, dust particles and foreign gases all exert a marked effect upon the production of ozone. The effect of these factors is not directly upon the ozone, but rather directly upon the character of the brush discharge which produces the ozone. A relative humidity of only 25 per cent at a dry bulb temperature of 20 deg. cent. will decrease the ozonizing power of the brush discharge from 30 to 40 per cent.

Moist air has a physical effect on the electrons, a chemical effect on the ozone that is produced and further, on precipitation, causes changes in the character of the brush discharge.

C. T. R. Wilson's[5] classical work demonstrated that ions act as nuclei for the condensation of water vapors, and he succeeded in photographing the trails of beta particles (electrons), by passing them through humid air, the camera recording the drops of water condensed upon the electrons.

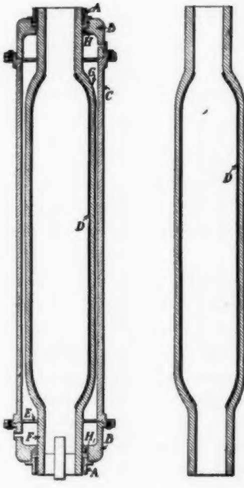
The electric current passing from one electrode to another in an ozonizer consists of a stream of electrons possessing considerable kinetic energy. Ozone is formed by

[5] Scientific Memoir of C. T. R. Wilson.

the action of these migrating electrons on the oxygen of the air. Molecular oxygen is split up and ionization is accomplished through collision between the migrating electrons and atomic electrons of the oxygen. A regrouping is thus effected and ozone is one of the results. It is necessary for the migrating electrons to possess sufficient kinetic energy to ionize oxygen, but it is not desirable for the electrons to possess sufficient energy to ionize the nitrogen present. Oxygen is a good electron trap and much less energy is required to ionize it than required to ionize nitrogen. This means that the energy possessed by the migrating electrons must be maintained within well defined limits.

When air containing moisture is used, the condensation of water on the electrons reduces their effective energy or power of ionizing oxygen, hence the production of ozone is reduced. When ordinary air from the atmosphere, which varies widely

FIG. 5. CROSS-SECTION OF OZONE GENERATOR



in vapor content, is used for ozonizing, it is impossible to obtain anything but widely varying ozone production. This variation will cover a range of from 10 to 70 per cent of the yield that would be obtained with dry air, and there is positively no means of detecting this variation, or recording it, except by an actual chemical analysis of the ozonized air produced by the ozonizer.

The water condensed by the electrons will be lodged on the dielectric or electrode of the ozonizer. This will produce edge effects which gives rise to sparking. T. Lowry[6] conducted an interesting research on the effect of the brush and spark discharges on air and found that under experimental conditions neither discharge gives rise to oxides of nitrogen when the air is dry. How ever, when dry air was passed through the two discharges in series, or the air from each discharge subsequently mixed, oxides of nitrogen were produced. Lowry came to the conclusion that the spark discharge produced an active variety of nitrogen that was easily oxidized by ozone. Lowry's findings have been proven in practice, since an ozonizer

[6] T. Lowry, *Journal Chemical Society*, **101**, 1152 (1912).

having both sparking and brush discharges occurring simultaneously will produce oxides of nitrogen from dry air. This conclusively demonstrates the necessity of avoiding edge effects in an ozonizer.

Practice has also revealed that an ozonizer free of sparking discharges, so far as this factor can be controlled by design, will produce oxides of nitrogen when operated with humid air. Here the question arises as to whether the precipitated moisture causes sparking, which alone is the cause of nitrogen being activated, or whether water, or perhaps its vapor, acts as a catalyst. Experiment and good reason would indicate that both contribute to the effect. Water and water vapors are instrumental in promoting many chemical reactions and while experiment shows that water does not promote reaction between ozone and nitrogen subsequent to issue from the generator, it does not necessarily follow that this holds for the conditions existing in the field of the discharge.

The net results of ozonizing humid air are then:

(*a*) A decrease in the ozone production of from 30 to 90 per cent
(*b*) The production of oxides of nitrogen
(*c*) Making it impossible to meter ozone by convenient electrical methods.

Dust

Dust in the air passed through an ozonizer favors the passage of sparks, and, as has been seen, sparking discharges are active in the fixation of nitrogen. The spark discharge is much hotter than the brush discharge and the thermal effects of sparking undoubtedly have a great deal to do with the effect on the nitrogen of the air.

Foreign gases such as sulphur dioxide, chlorine, etc., appreciably decrease the amount of ozone produced.

In the design of an ozonizer the first consideration should be to effectively shut out the outside atmosphere and utilize only a small quantity of perfectly conditioned air from which to produce relatively high concentrations of ozone. The evils of sparking have been mentioned. Edges or points always favor the passage of sparks, hence, an ozonizer should be so designed as to eliminate the possibility of edge or point discharges. The boundaries of the electrodes require special treatment to avoid sparking discharges. Fig. 5 illustrates a patented means of doing this. This figure is a cross section of a Type-T U. S. ozone generator. The heavy black lining of the cross hatched central member, represents the metallic lining which serves as the central electrode. It will be noted that the shape of this central member is such as to form cone frusta at each end. In this way the polar distance (distance between the two electrodes) is gradually increased, so that the discharge gradually fades out. Thus discharge from abrupt edges is avoided. The cross hatched portion of the central member is of glass and serves as the solid dielectric. It will also be noted that the glass begins to gradually thicken along the radii, thus the dielectric strength is increased, serving to further diminish the activity of the discharge near the electrode ends. The outer cross hatched portion is the aluminum casing of the generator, and serves as the outer electrode. It is very important that the radius employed be of just the proper degree of taper, and this has been worked out by careful experiment.

Fig. 6 shows a Type B assembly employing the Type-T ozone generators, and equipped with a porolyte dehydrator and air filter. This unit has a capacity of 50 grams of ozone per hour, when operating on 60 cycle current. In the illustration, one side panel has been removed to show the arrangement of the interior. The

porolyte dehydrator is shown at the right of the ozonizer. The motor-driven blower is situated behind the ozonizer and is not visible in the photograph.

As dry air is a prerequisite of ozone production, a few words will be devoted to the porolyte dehydrator, which is in a measure, an unique apparatus for the production of dry air. Porolyte is a name which has been given to a granular silica substance, containing an enormous number of exceedingly minute pores. The porolyte particles are of such size as to pass a Number 10 screen, but be retained on a Number 12 screen, and the air to be dried is blown through a quantity of the porolyte contained in a suitable chamber. The removal of the water vapor is purely

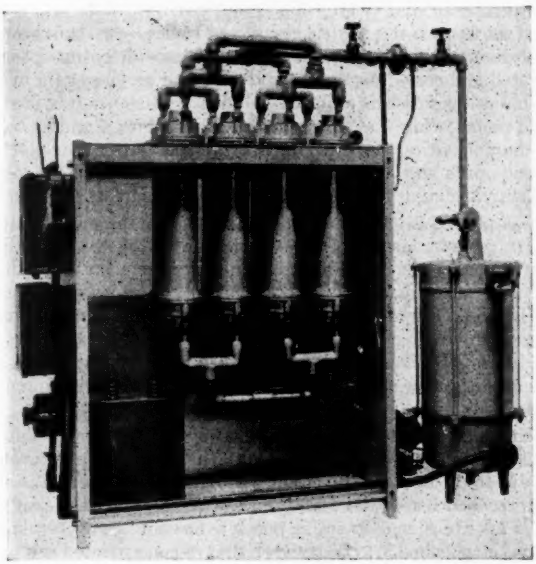


FIG. 6. OZONE GENERATORS, POROLYTE DEHYDRATOR AND AIR FILTER

physical in character, and since porolyte is exceedingly inert chemically, its life is practically indefinite.

Porolyte presents to the air an enormous surface that is exceedingly dry, when the porolyte is in the active state. The water vapor is first absorbed on the surface of the porolyte and this absorbed water film is taken up by the pores, by capillary attraction, this process being continued until the pores become saturated when the drying operation ceases. Porolyte is hygroscopic but is not deliquescent, in fact saturated porolyte is not sensibly moist to the touch. Since the absorption of moisture is physical, deliquescence is impossible, hence porolyte cannot be injured by exposure to moisture in the saturated state. If active porolyte is plunged into water, great heat is evolved and the granules are destroyed by decrepitation.

The value of porolyte as a desiccating medium rests in its ability to give up the water that has been condensed in its pores, on heating. To cause the condensed

water to escape it is simply necessary to raise the temperature of the water within the pores, to a point where the vapor tension is greater than the capillary force. Or, looking at it from the view point of equilibrium, the hot air has a greater capacity for vapor than the porolyte and the equilibrium is shifted in favor of the air. Heat can best be supplied to the porolyte by passing hot air through it. This heat is rapidly absorbed by the water in changing from the liquid to the vapor phase, and in consequence the air issues from the bed saturated at a temperature of about 50 deg. cent. After all of the vapors have been driven from the porolyte, the temperature of the issuing air will abruptly rise to nearly its initial temperature, which is about 125 deg. cent. This affords excellent means of automatically stopping the activating operation immediately after it is completed. A thermostat is placed in the outlet air and set to open the electric circuit at 115 deg. cent. thus stopping the activating operation. For intermittent service a single dehydrator serves, since it can be activated and cooled over night and ready for service again in the morning. Where continuous operation is required two dehydrators are provided, one can be activated and cooled, while the other is in operation. Figs. 7 and 8 give an excellent idea of the characteristics of the drying operations. Fig. 8 was plotted from observations made on a size 3 dehydrator which is rated at 3 cu. ft. per min.; at 7 grains of moisture per cubic foot, this dehydrator will completely dry air for 72 hours.

Figure 7 was plotted from observations made on the same dehydrator but operated at 7 cu. ft. per min. instead of 3 as was the case in Fig. 8, and the drying operation was started before the porolyte had been cooled from the activating operation. The figures on the graph denote the temperature of the air issuing from the porolyte at the time the observation was made. It will be noted that 4 hours were required to cool the porolyte to room temperature and the amount of vapor escaping absorption was fairly proportionate to the temperature. The drying of the air was complete for 26 hours at a flow of 233 per cent. At 30 hours the total quantity of water absorbed was 15 lb.

The capacity of this size unit is 20 lb. of water. However, in this experiment, the air flow was increased 133 per cent, hence, the outlet temperature was appreciably raised, due to the heat of absorption, and equilibrium due to the elevated temperature was reached before the porolyte was fully saturated. Accurate determination of the vapor tension over active porolyte has not been determined, however, experiment has established it to be between that of concentrated sulphuric acid and phosphorous pentoxide. The vapor pressure over concentrated sulphuric acid at 50 deg. cent. is approximately 0.007 mm. According to Morley[7] phosphorous pentoxide leaves not more than 1 mg. of water in 40,000 liters of air. Since it takes phosphorous pentoxide to detect the vapor left in air after treatment with porolyte we can readily see that porolyte produces perfectly dry air.

Ozone in Water Purification

Ozone has, perhaps, been more widely applied to water purification than to any other commercial process. There are in operation in the United States, Canada and Mexico, more than a thousand individual ozone water purifying plants, purifying millions of gallons of water a day. On Continental Europe the water supply of such cities as Paris, Madrid, Vienna and numerous provincial cities and towns of France and Germany, is purified by ozone. Ozone is an ideal water purifying agent, since it not only destroys germ life, but oxidizes a considerable portion of the soluble organic matter, which cannot be removed by filtration. Odors and tastes,

[7] *Amer. Journal Sci.*, **30**, 441 (1885).

arising from an organic origin, are readily destroyed by ozone leaving the water sparkling and palatable.

The purifying principles of ozone are purely those of oxidation. A bacterium is composed of hydrogen, carbon and nitrogen. Ozone combines with these substances to produce carbon dioxide, water and nitrates. The soluble organic matter found in water supplies is generally of a vegetable origin, however, frequently protein matter is found in water supplies. These organic substances are such as are readily disintegrated by hydrolysis, and the products of disintegration are highly tasty and odoriferous. The chlorination of such water frequently produces chlorine substi-

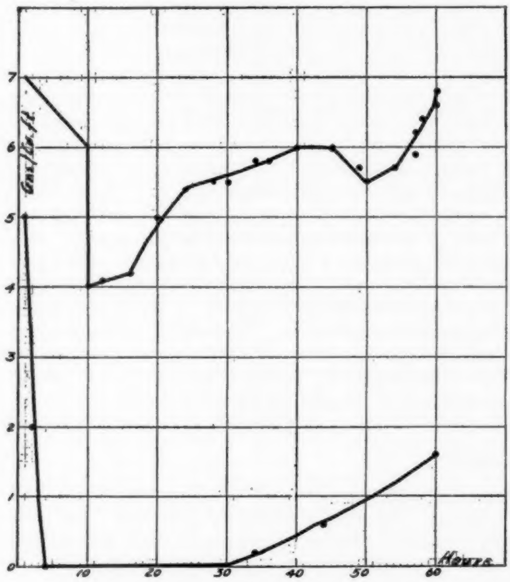


FIG. 7. CHART SHOWING DRYING OPERATION DEHYDRATOR OPERATING AT 7 CU. FT. PER MIN.

tution and addition products, that are much more objectionable than the original substance. Ozone not only eliminates these substances before chlorination, but can be successfully used to remove the chlorinated product after the chlorine treatment. Ozone water-purifying equipment is frequently installed in beverage and food product plants for the specific purpose of removing tastes and odors arising from the chlorine treatment of the municipal supply. The ozone treatment is particularly valuable where water contains chlorinated phenols.

The cost of water purification with ozone is very reasonable, ranging from 100 to 300 watt hours per 1000 gal., depending upon the capacity of the plant. The quality of water produced by ozone treatment is of the highest order. Ozone in solution in concentrations as low as one part per million is immediately destructive to all pathogenic organisms, and in concentrations up to four parts per million is destruc-

tive towards the more common spore bearing species. Due to the low solubility of ozone at atmospheric temperatures and pressure, it is impossible to have a residue of the purifying agent in the water after purification. Objectionable tastes and odors are eliminated, and the low oxidation organic matter is oxidized to innocuous gases such as CO_2 and nitrogen.

The sanitation of a swimming pool offers many difficulties. It is the one application of water purification where the purity of the water over a protracted period of time is of importance. At the present cost of water it is impractical to fill a pool daily, and even if this is practiced the sanitation of the pool will be below par. A swimming pool should be supplied continuously with pure water, in quantities equal to one-tenth the capacity of the pool, per hour. The only way to do this economically is by recirculation and purification. Effective filtration is difficult since the continual addition of alum to the water soon exhausts its temporary hardness and there is nothing left to precipitate the aluminum hydroxide unless chemical treatment be resorted to. On a small scale, with the process in the charge of a janitor, chemical treatment of the water is always unsatisfactory. In addition to the bacteria introduced into the water by the bathers, considerable organic secretions from the body find way into the pool. This calls for some means of oxidizing this class of impurities and ozone fills this want very completely indeed. The action of ozone on such substances as urea, for an instance, is to produce carbon dioxide, water and nitrogen. There is an ozonizer operating on a pool in Michigan that has been in operation for more than a year. Analyses made of the pool water reveals it to be of a higher order of purity than the city supply. This water has not been changed since the ozonizer was placed in operation, with the exception of the make-up water added to replace evaporation. The primary requirements of an ozonizer for water purification are, a constant quantity of ozone from the ozone generators, and an adequate mixing device for mixing the ozone with the water. The results obtained by a water ozonizer may be readily ascertained by analyses of the finished product, and the process is thoroughly reliable and automatic in every respect.

Ventilation Applications

Ozone finds valuable application in the art of ventilation. The quantity of ozone required being of the order of from 0.01 to 0.10 part per million by volume; for general ventilation, and not exceeding one part per million in the storage of perishable products. One-tenth part per million by volume of ozone amounts to 5.7 mgs. per 1000 cu. ft., so it is readily seen that ozone is an exceedingly cheap reagent for air purification.

If is frequently claimed that ozone is an active aerial germicide. This statement is not wholly without truth, but as enunciated it is often misleading, regardless of the intent behind the statement. Compared with formaldehyde, sulphur dioxide and other gases frequently used for disinfecting, ozone rightfully holds first place; but the concentration of ozone necessary for germicidal action in air having a normal vapor content, is far beyond the respirable limits of this gas. In fact one per cent ozone is necessary for strong germicidal action in air, whereas the other disinfectants generally employed, require concentrations of four to ten per cent. The great value of ozone in ventilation resides in its powers to destroy odors and impart to the air a certain freshness that is suggestive of outdoor air.

A consideration of the facts in the case reveals very forcibly the value of ozone in ventilation. Some of us may unthinkingly presume that our atmosphere reaches out to the utmost star and that there is an unlimited supply of fresh air from which to draw, when we have polluted and done with the air that we cast out of our ex-

haust systems. Such, however, is far from the case. As a matter of fact nature practices recirculation on a gigantic scale, just as we would like to practice it on a more minute scale. The earth's troposphere extends something less than seven miles in height. There is practically no interchange of gases between the troposphere below and the stratosphere above. A circulation of the earth's atmosphere is effected by convection, warm air arises at the equator and descends at the poles, thus we have a good air movement, but it is the same air that blows out of the north that we at some previous day sent down to the south after we had done with it. Obviously, nature must provide some means of purifying this air, since contamination is so prolific, simple dilution is entirely out of the question. Nature has provided the green leaf pigment to maintain our oxygen-CO_2 balance. The rains pre-

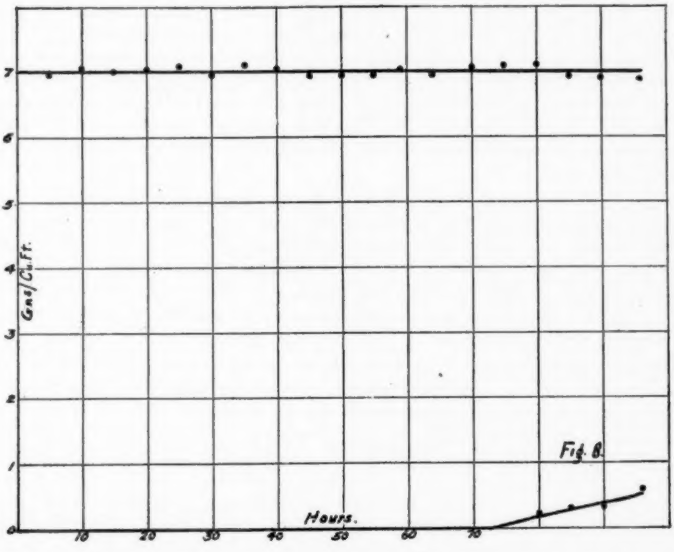


FIG. 8. CHARACTERISTICS OF DRYING OPERATION DEHYDRATOR OPERATING AT 3 CU. FT. PER MIN.

cipitate the solid matter that is cast into the air and oxidation accounts for the disintegration of the complex substances that the air is called upon to dispose of.

The oxygen of nature is very active; however, it is found that if it is housed or bottled up it loses much of its activity. Then there must be something in nature that makes oxygen active. These activating principles of nature are found in the short wave length emanations of sunlight. In radioactive changes that are continually going on in the earth, in the evaporating of water under certain conditions in nature, and in the intense electrical discharges of lightning, etc. The activating principle of these phenomena is ionization. Ozone is a product of ionization and in air that has been ozonized, the accompanying oxygen is largely ionized.

The question exists in the writer's mind as to whether ozone is the sole beneficial

agent in the air from an ozonizer. It is well known that oxidations can be accomplished with ozone greatly in excess of the ozone present. More on this point will be discussed when the treatment of linseed oil with ozone is touched upon. We are prone to look upon this phenomenon as one of catalysts. But what are catalysts? In the present status of our knowledge it is frequently an answer rather than an explanation. To me there is small doubt but that this extra oxidation, so characteristic of ozonized air, is accounted for by the high state of ionization possessed by the oxygen accompanying the ozone and that this oxygen is capable of activity denied to oxygen in its unionized state. The air of our cities is called upon to bear so great a burden that there is small wonder that the edge is knocked off of it. The problem that confronts the ventilating engineer is the production of a fresh air condition in buildings. The cards are stacked against him from the start. There is only a source of questionable air for him to draw from, and this air must be handled with all manner of mechanical contrivances that adds nothing to its zest. The fresh air of the rural districts is nothing other than the rejected air of our cities that nature has purified in her own laboratory. Is it not a very logical step to simulate nature, and purify the doubtful air of our cities with exactly the same method that she employs? An ozonizer places the tools to do this, into the hands of the ventilating engineer, as any one will find out who will take the trouble to give the matter the study which it merits.

Previously it has been remarked that ozone, in respirable concentrations, is not germicidal. However, in cold storage ozone is successfully used to prevent the growth of fungi. On the surface this may seem a paradox, but such is not the case. A substance may be an antiseptic without being a germicide. Formaldehyde is an antiseptic in 1 to 50,000 dilutions and is not germicidal until the concentration reaches 1 to 10. An antiseptic is, as its name implies, merely an inhibiting agent, and not necessarily destructive.

Ozone, in concentrations of $^1/_2$ to one part per million is decidedly inhibitory towards nearly all forms of microorganisms. Some years ago the writer conducted some extensive experiments to ascertain the germicidal powers of ozone in air. About the first experiment was to subject a contaminated atmosphere to treatment with ozone, and filter the air for the organisms and subject the filtrate to incubation on culture media in the usual way. It was soon found that ozone was not germicidal until concentrations of 13 to 14 mgs. per liter were reached. The atmosphere which I worked with was confined in a wooden chamber and as I had heavily inoculated this chamber with mold spores, I was surprised to find no mold growth on the sides of the chamber and on bits of cork that were lying about in the chamber during the experiments. Conditions were ideal for the development of mold. The air was warm, moist and free from violent motion. The chamber had the strong light of day excluded from it, yet no molds developed. This suggested an idea that was exploited in a new set of experiments.

Bread and leather were inoculated with mold spores and placed in a glass chamber that was excluded from the direct light. Ozonized air was lead through this chamber in such a way as to avoid draughts but give a good mixing of the ozonized air. A blank experiment was mounted and conducted along side of the one with the ozone. In a few days a good mycelium growth was noticed in the blank chamber but no development in the one with the ozone. The experiment was continued for several weeks, by which time there was a prolific growth in the blank chamber and still no growth in the one with the ozone. The contents in the two chambers were then reversed and in a few more days a growth again appeared in the blank chamber on the specimens that had previously been in the ozone chamber, and no appreci-

able arresting of the growth was noticed on the specimens then in the ozone chamber. This indicated that ozone was inhibitory but not destructive to the fungi. These experiments were then conducted over six months, with air having a relative humidity of from 50 per cent to saturated, over a range of temperatures of nearly 0 deg. cent. to 40 deg. cent. When the concentration of ozone was approximately $^1/_2$ part per million, or better, the inhibitation was perfect, but on removal of the specimens from the atmosphere of ozone and followed by careful incubation, developments were obtained. This conclusively shows the action of ozone in low concentrations and that it is decidedly inhibitory towards fungi is undoubtable. This property gives to ozone an enormous value in cold storage. Coupled with this, the deodorizing properties of ozone makes it a necessity in cold storage where the problem of keeping fresh products is daily increasing.

Eggs have been carried at a relative humidity of 88 to 90 per cent and mold development inhibited with the use of ozone. Cheese, cabbage and onions have been stored in the same room and taste transfer avoided by the use of ozone. Rooms that have been used for odoriferous products can be deodorized and made sweet in a few hours time without a loss of temperature. Ozone has manifold applications in cold storage, and splendid results are being obtained in practice with this reagent every day.

Bleaching Operations

Ozone has been successfully used in many bleaching operations. In a recently completed plant for large scale operations on the bleaching of bees wax, the procedure is about as follows: The wax was heated in steam-jacketed iron kettles to its point of liquefaction. Ozonized air, at a concentration of one to three milligrame per liter is then blown through the wax from a perforated manifold located in th- bottom of the kettle. Bees wax can be bleached to a very good color at a very reas sonable cost, since very low concentration ozone is best for this purpose. Low concentration ozone with prolonged treatment yields better results than higher concentration ozone with corresponding shorter exposure.

The waxes that have been worked with are the Zanzibar, Egyptian and African waxes. Yellow Zanzibar wax can be bleached to a very satisfactory color with from one and one-half to two grams of ozone per pound of wax. The Egyptian waxes require from two to three grams of ozone per pound of wax, while the African wax does not respond so readily to the ozone treatment and a good white color could not be obtained with this wax at any treatment given it in our work. A two and one-half kw. 1000 cycle ozone plant will produce about a ton of bleached wax per 24 hours.

Effect on Linseed Oil

The effects of ozone on linseed oil are very interesting, and may be divided into two (2) distinct and independent classes: (*a*) bleaching and (*b*) an oxidizing effect.

When the temperature of the oil, during ozonizing, is maintained below 25 deg. cent. the action is that of bleaching; when the temperature of the oil is raised above 35 deg. cent. a marked oxidation results.

In order to obtain the best results in the bleaching of linseed oil, this oil was first refined with alkali and the alkali soaps settled out and the oil carefully washed with water.

The refined oil was subjected to treatment with ozone in lead-lined kettles equipped with a propeller type agitator. The ozone was led in directly below the

propeller, which served to form a very good oil-air emulsion that permits of a permeation of the oil body more thoroughly than when the ozonized air was introduced through perforated pipes, with no provisions for stirring, or the ordinary rake type stirrer used. Ozonized air (0.5 per cent ozone in air) in quantities of 5 grams of ozone per gallon of oil was allowed to act on the oil at temperatures varying between 15 and 20 deg. cent. for five hours. At the expiration of this time the oil will have acquired a pale straw color. In films 1 mm. thick, this oil is absolutely colorless, the color not being readily noticeable until thickness approaching 2 cms. are obtained.

During the course of our work, we encountered several samples of oil that did not bleach at once. Laboratory data revealed that these samples had a very low iodine number, accompanied by high acidity. These oils, after the usual five hours, treatment, had not appreciably changed color. Further ozonizing was not attempted but they were exposed to diffused sunlight. In 10 days' time they bleached to the extent of the other samples. Some after bleaching were noticed with all of the samples worked on, and their drying properties were enhanced.

Further laboratory investigations revealed that an oil that has been clarified, by heating to the break, will not bleach well, nor will a long tanked oil give good results. A relatively cold pressed oil from the last year's crop of seeds is by far the best material for ozonation.

Our work has been stimulated considerably by inquiries from both varnish makers and linoleum industry, and for the latter we took up the question of oxidizing linseed oil.

Ordinary raw oil of commerce was used in this work, and the following is an epitome of the results obtained:

Time of Ozonizing[3]	Drying Time
1 hour	30 hours
2 hours	26 hours
4 hours	21 hours
6 hours	18 hours
8 hours	10 hours
12 hours	Did not dry

It is interesting to note that the last oil did not dry.

The above samples were treated with 15 grams of ozone, per gallon of oil, per hour, the concentration of ozone being 0.5 per cent. All of these oils acquired a peculiar cucumber-like odor, that is very pleasing, and this, we have noted, is characteristic of all ozonized vegetable oils. The viscosity of the oil was appreciably increased, but a portion of this was lost on standing.

An analysis was made of several of the ozonized samples for the same raw oil with the following results:

TABLE 1. ANALYSIS OF RESULTS

	Carbon	Hydrogen	Oxygen
Raw Linseed Oil	87.45%	7.15%	5.4%
Oil treated 8 hours, O_3-air	83.95%	6.86%	9.2%
Oil treated 8 hours, O_3-O_2	79.87%	6.52%	13.6%
Oil treated 12 hours, O_3-O_2	67.65%	5.53%	26.8%

[3] No metallic dryers were used.

TABLE 2

	Iodine Value	Oxygen	Added Oxygen
Raw Oil	180	5.4%	nil
Oil treated 8 hours, O_3-air	150	9.2%	3.8%
Oil treated 8 hours, O_3-O_2	115	13.6%	8.2%
Oil treated 12 hours, O_3-O_2	10	26.8%	21.4%

Excellent results have been obtained with all grades of linseed oil in so far as the effect of oxidation is concerned, but to bleach an oil economically with ozone it will be necessary to maintain some method of control over the raw materials, and the history of the oil should be known when possible.

We have also done a great deal of work on the oxidation of linseed oil, and linoxyn can be economically formed with the use of ozone. The best results are obtained by pre-ozonizing the oil and then putting it through the usual so-called mechanical process of producing linoxyn.

A very promising use of ozone is found in the production of vanillin from isoeugenol. Isoeugenol is dissolved in absolute ether and a slight excess of 0.5 per cent ozone lead in at room temperature. The ozonized ethereal solution is carefully concentrated to about one-third of its volume and the ozonide reduced with acetic acid and zinc. The excess zinc is filtered off and the solution neutralized with calcium carbonate. The solution is then washed with water and then agitated with sodium bisulfite. The sulfite addition product comes down readily, is decomposed with sodium hydroxide and the vanillin precipitated by careful neutralization with sulphuric acid. The process is 75 per cent efficient and $^3/_{10}$ lb. of ozone will produce $^4/_{10}$ lb. of vanillin from 1 lb. of isoeugenol.

In the literature it is found that a process involving the use of acetylisoeugenol is given some prominence, but while we were able to demonstrate the formation of an ozonide, when attempting to duplicate the work, we were unsuccessful in precipitating the acetyl vanillin.

Ozone has also been widely used in the preparation of synthetic perfumes, particularly the phenolic aldehydes.

Anisaldehyde (*p*-methoxy benzaldehyde) may be prepared from methyl allylphenol (anethol) ozone, and piperonal (methylene-protocatechuic aldehyde) or synthetic heliotrope may be prepared from safrol.

In investigating the structure of various complex organic compounds ozone offers a very convenient tool. The structure of compounds containing ethylene linkages may be elucidated by preparing the ozonide and subsequently decomposing it. Harries (*Ber.* 33, 839, 842, 2708, 3431 (1904) et. seq.) has made considerable use of this property of ozone.

The position of the unsaturated linkage in oelic acid was established in this manner, it being located between the atoms C_9H_{10}.

Ozone ruptures the unsaturated compounds at the double bond, converting them into aldehydes and ketones. In the absence of water, however, a direct addition of ozone to the double bond will occur, thus forming ozonides, which on the subsequent addition of water, undergo decomposition into ketones and H_2O_2.

The ozonides are colorless, viscid, oil substances that are highly explosive and possessed of a penetrating odor. They behave like powerful oxidizing agents, liberate iodine from KI and react with $KMnO_4$. Their most characteristic reaction is perhaps their action on a photographic plate.

Considerable work has been done with ozone on rubber, which it attacks very energetically, destroying it by depolymerization.

DISCUSSION

H. R. LINN: Do you have any tests where ozone has been used in hospitals in connection with pneumonia cases or other cases of lung disorder?

F. E. HARTMAN: I believe that is outside of our scope. When you are going into physiological reactions, it doesn't behoove the layman to take it up. That is an investigation that lies thoroughly within the field of the physician and it is impossible to make any guarantees or any claims on therapeutic or physiological reactions of ozone until your data is comprehensive and covers observations made over a long period of time and on a number of subjects, because we have personal differences to take into consideration and negative result means nothing. There is quite a literature on the therapeutic uses of ozone, and I believe the first prominence given to ozone in the treatment of respirative diseases was given by a physician in Boston. I can't recall his name but I would be very glad to give you the bibliography of that literature if you wish.

No. 714

THE USE OF OWENS' JET DUST COUNTER AND OF ELECTRIC PRECIPITATION IN THE DETERMINATION OF DUSTS, FUMES, AND SMOKES IN AIR

PHILIP DRINKER[1] (MEMBER)

ROBERT M. THOMPSON[2] (NON-MEMBER)

BOSTON, MASS.

Introduction

THE quantitative determination of dusts, fumes, and smokes in air presents a problem of interest and practical importance to the ventilating engineer, hygienist and the physician.

If the air of a building is contaminated with abnormal amounts of dusts, fumes, and smokes, the engineer requires quantitative determinations in order to compute his ventilation requirements. If increased ventilation is applied, he must determine the extent of the improvement produced. If the impurities constitute a menace to health, the physician must know the amount in the air breathed, the size of the particles, their number and their chemical and physical nature. Even with these data available, the physician's opinion may still be uncertain unless he has present reliable statistics, post mortem findings, or positive evidence from clinical examination of patients exposed. Without such quantitative determinations neither engineer nor physician can draw up standards of permissible amounts of dusts, fumes, and smokes in air.

Distinction between Dusts, Fumes, and Smokes

The size of the particles is the chief factor in governing the rates at which dusts, fumes, and smokes settle out in still air. Definite physical laws have been evolved from which the theoretical rate of settling of any particle in still air may be calculated. A classification based on this principle was suggested by Gibbs,[3] who distinguished between dusts, fumes, or clouds, and smokes, on the basis of particle size. The objection to his classification is that still air conditions are almost never encountered, so that classifications based solely upon particle size and still air condi-

---

[1,2] Instructor, Department of Ventilation & Illumination, Harvard School of Public Health.
[3] Gibbs, W. E.: "The Industrial Treatment of Fumes, and Dusty Gases," *Journal Soc. Chem. Indus.*, **41**, 189 T (1922).

Presented at the Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Boston, January, 1925.

tions are necessarily impracticable. While a highly accurate classification is probably unobtainable, the following is suggested as coming nearer to conditions as they are met in practical work:

1. DUSTS—Particles of 150 to 1 micron in diameter. Such particles are thrown into the air by mechanical agencies such as grinding, crushing, drilling, and blasting.

2. FUMES—Particles of 0.2 to 1 micron, resulting from reactions such as

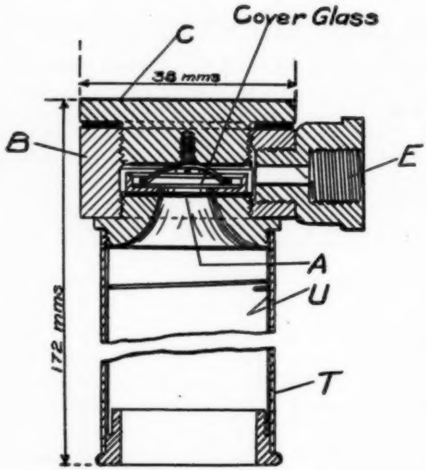


FIG. 1. VERTICAL SECTION OF OWEN'S JET DUST COUNTER (Courtesy of *Journal of Industrial Hygiene*)

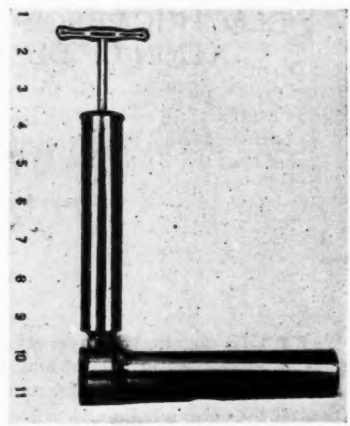


FIG. 2. OWENS' JET DUST COUNTER SHOWING SCALE IN INCHES

distillation, complete and incomplete oxidation of metal fumes, and purely chemical reactions.

Examples: Ammonium chloride, lead and mercury fumes, zinc oxide, sulphuric acid mist.

3. SMOKES—Particles less than 0.3 microns, usually resulting from incomplete combustion of carbonaceous material such as coal, oil, tar, tobacco, etc.

Whether formed by chemical reactions, or present in the air originally, moisture often exerts an appreciable effect on the formation of fume and smoke particles, tending usually to increase the visibility of the particles by condensing upon them as nuclei. It may exert a similar sort of effect, but to a less degree, on dust clouds.

The line of demarcation between classes (1) and (2), or between (2) and (3) is neither sharp nor positive, and particle size alone rarely gives enough facts to make the classification distinct.

In general the particles in these three classes increase in uniformity with decrease in size. The size of dust particles covers a very wide range, fume particles are smaller and more uniform, while smokes show the greatest uniformity and are usually too small to be measured under the ordinary microscope. Substances like lamp black, zinc oxide, gas black, and lead fume, on collection and reblowing into

the air often behave like dusts. If such particles, blown into the air mechanically, are then examined under the microscope, they will be found to be flocculated and agglomerated into aggregates consisting of large numbers of fume or smoke particles. Therefore the method of which the dust, fume, or smoke cloud was generated must not be ignored by either engineer or physician.

In practical work the problem is often complicated by mixtures, in which case, such proposed distinctions are difficult to apply. In mixed dusts the engineer is generally concerned with the total amount of the dust, while the hygienist or physician may be interested in but one of its many constituent part. However, fine distinctions need not interfere with the selection of the method chosen for quantitative determinations.

Methods of Determining Dusts, Fumes, and Smokes

In a review of the literature of methods for determining dust, Drinker, Thomson and Fitchet[4] discussed five classes. For one of which, counting, impingement may be substituted. The five classes then become:

(*a*) Settling
(*b*) Impingement
(*c*) Filtration
(*d*) Scrubbing
(*e*) Electric Precipitation

Of these five methods, electric precipitation and impingement may often be applied when other methods lack the required accuracy. Also, electric precipitation and impingement are applicable to the determination of dusts, fumes, and smokes, while the other three methods (settling, filtration, and scrubbing) are best adapted to the determination of dusts, and are of doubtful use against fumes or smokes.

Since electric precipitation and Owens' impingement method may often be applied where other methods, applicable under less exacting circumstances, fail to achieve the required accuracy, it was thought that a discussion of these two methods would interest the members of this Society.

Impingement

If a stream of dusty air is driven against a glass slide coated with a sticky substance, such as vaseline, the particles are caught on the vaseline at efficiencies which vary with the velocity of the air jet and the nearness of the glass slide to the jet.[5],[6] If the dusty air is humidified before discharging against the slide, and no vaseline is used, it has been shown by Owens[7] that better results are obtained.

The general principle has been modified in Greenburg and Smith's[8] method. The impinging plate and nozzle, through which the dusty air passes, are placed under water, and the dust caught on the plate is continually washed off into the containing vessel. Kotze's[9] new "hydrokonimeter" is based on the same principle, but Kotze takes a sample of about 50 cc. of air, while Greenburg and Smith take several cubic feet.

---

[4] Drinker, P., Thomson, R. M., and Fitchet, S. M.: "Atmospheric Particulate Matter: I Dust with a New Apparatus for its Determination," *Journal of Industrial Hygiene*, **5**, 19, 62 (1923–24).

[5] Hill, E. V.: "Ventilating Division of the Health Department, Chicago, Ill.," JOURNAL AM. SOC. HEAT. VENT. ENG., **19**, 421 (1913).

[6] Innes, J.: "The Estimation of Injurious Dust in Mine Air by the Kotze Konimeter," *Journal Chem. Met. Min. Soc., S. Africa*, **18**, 199 (1918); **19**, 132 (1919).

[7] Owens, J. S.: "Jet Dust Counting Apparatus," *Journal of Industrial Hygiene*, **4**, 522 (1922–23)

[8] Greenburg, L., and Smith G. W.: "A New Instrument for Sampling Aerial Dust," *U. S. Bur. Mines, Rep. Investigations*, **2392** (1922).

[9] Kotze, Sir R. N.: "A New Method of Dust Determination," *Journal of the Chem. Met. and Min. Soc. of S. Africa*, **24**, No. 1, July (1923).

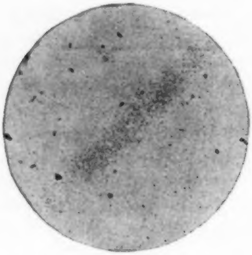


Fig. 3. 50 cc. by Owens' Counter. Taken at 9.50 p.m. Plant in Full Operation

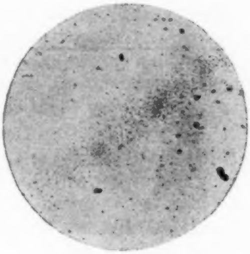


Fig. 4. 50 cc. at 10.55 p.m.

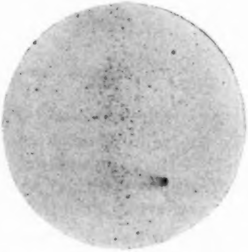


Fig. 5. 50 cc. at 11.10 p.m.

Fig. 6. 100 cc. at 11.50 p.m.

Fig. 7. Ammonium Chloride Crystals from Fume Caught by Owens' Counter. (Approximately 125 d. Magnification)

Figs. 3–6. Settling Test of Dust in Large Room Where Ore is Crushed and Dried. (Approximately 125d. magnification.) Plant Shut Down at 10.50 p.m. and Source of Dust Disturbance Ceased

Owen's Jet Dust Counter

The principle on which the functioning of this instrument is based is given by Owens,[7] as follows: "A high velocity jet of air is caused to strike a microscope cover glass; the effect of this high velocity is to bring about a fall of pressure in the jet, accompanying which, and resulting from it, is a corresponding fall of temperature. This in turn causes a condensation of the moisture in the air upon the dust particles, which are thus projected wet against the cover glass, and, as the water evaporates, are left behind adhering to the glass."

As already stated, Owens found that if the sample of air was retained for a moment in a humidifying chamber before being drawn through the orifice, the condensation effect of the added moisture on the dust particles increased the efficiency of the apparatus. Consequently the humidifying chamber is an important part of the instrument.

The temperature of the cover slip against which the jet impinges is always appreciably higher than that of the jet. Therefore, in dry atmospheres, the moisture condensed by the jet action is evaporated practically instantaneously. In very humid atmospheres the drying effect is retarded so that the cover slip may show moisture for some moments after the sample has been taken.

Constructional Details. The apparatus, Figs. 1 and 2, consists in a dampening tube, *T*, lined inside with a moistened blotter, *U*, an orifice, *A*, with a trumpet shaped approach, a sleeve, *B*, to which the suction pump is screwed at *E*, and removable head, *C*, for holding the microscope cover slip in place close to the exhaust side of the orifice, and the 50 cc. (28,300 cc. = 1 cu. ft. approximately) suction pump which screws in at *E*. The apparatus, with the exception of the orifice is nickeled inside and out and joints are provided with leather gaskets where needed.

Taking of Samples. In sampling dust, fume or smoke with this instrument, the blotter in the humidifying chamber is first moistened with water, and a number of strokes taken to displace the residual air with the air to be sampled. The removable head, *C*, is then unscrewed, a cover slip dropped into place, and the head screwed home. One or more stokes of the pump are then taken, the head *C*, unscrewed, and the cover slip allowed to fall out with the dust side up. Usually a fine ribbon of about the dimensions of the orifice in then plainly visible, particularly if the cover slip is held up to the light. The cover slip is mounted on top of a flat ring fastened to a microscope slide by a special adhesive which softens under gentle heating. The ribbon of dust is then on the underside of the cover slip with a small air gap between the cover slip and the slide.

Before mounting, if there is difficulty in finding the dust ribbon under the microscope, breathing, on the dust side of the cover slip will usually render it visible. Owens points out that such difficultly visible ribbons can often be seen with dark ground illumination. If the air sampled is fairly pure, or the sample taken out of doors, it is often necessary to take several strokes with the pump, each stroke representing 50 cc. of air. Otherwise too few particles are present to give a visible ribbon.

Examination of Samples. Owens recommends counting the particles in a definite fraction of the ribbon, and for this purpose uses a net-ruled micrometer in the eye piece of the microscope. This procedure is simple, and the technique easy to acquire. However, counting is futile when the sample contains enough dust to cause the particles to pile up in layers. Unless such samples are photographed, they are of doubtful use in comparing degrees of dustiness.

Figs. 3-6 show a series of four dust samples taken in a large room in a plant where rock was being crushed, dried and transported by belt and bucket conveyors. The dust contained one rather toxic ingredient, identifiable under the microscope. It was desired to determine the rate at which dustiness in this plant subsided after the plant shut down, and also, whether the toxic ingredient settled out at a markedly different rate from the other ingredients of the dust. To determine these two points, a series of Owens' samples, which are practically instantaneous, were preferable to a series in which each member was taken over several minutes.

The particles in all of these samples could be counted, but the photographs show the ease with which differences in dustiness may be detected by photography. In Fig. 3 the count was a little uncertain because of the piling up of the particles. Samples from dustier atmospheres give even denser dust ribbons and are of little use unless photography is used.

The numerous particles outside the ribbon in Figs. 3 and 4 result from careless handling of the cover slip, and the slide on which the cover slip is mounted. In very dusty atmospheres it is difficult to mount the cover slips without getting stray dust on the slide. However, such dust is usually out of focus and need not affect the results.

Samples Taken in Moist Atmospheres.—If the per cent relative humidity is high in the air to be sampled, too much moisture is condensed on the particles, and those soluble in water dissolve on the cover slip, and crystallize out as the water evaporates. In the presence of sufficient moisture, soluble particles such as common salt, and ammonium chloride crystallize in patterns along the dust ribbon, as in Fig. 7. Such crystals often give a convenient method of estimating the constituents of a mixed dust or fume in air, but at the same time may give the observer incorrect ideas of the original state in which the particles occurred in the air.

Owens' apparatus was devised for taking samples for meteorological purposes, such as fog and smoke particles in the air of cities. Of necessity, an instrument destined for such work had to be very sensitive. As it is now manufactured, its usefulness in general dust problems would be increased if samples smaller than 50 cc. could be taken. Nevertheless its present advantages of accuracy, simplicity, cheapness and light weight are considerations not to be ignored.[10]

Electric Precipitation

If air containing dust, fume, or smoke is subjected to a high tension alternating air direct current discharge particles are precipitated in varying degrees of efficiency. This fact was noticed about a century ago, but received no successful commercial application until Cottrell,[11] in 1911 constructed his first plant for the recovery of sulphuric acid fume. From that date the development of the process which usually bears Cottrell's name, has been rapid, and the literature is replete with descriptions of large Cottrell plants all over the world for recovering dusts, fumes, and smokes formerly allowed to escape into the air.

Cottrell plants use high tension unidirectional or rectified, and not alternating current. The air to be treated passes through metal pipes down through the center of which are hung wires. The wire is made the negative and the pipe the

[10] The Owens Jet Dust Counter is made by C. F. Cassela & Co. Ltd., London, England.

[11] Cottrell, F. G.: "The Electrical Precipitation of Suspended Particles," *Journal Indus. and Eng. Chem.*, **3**, 542 (1911); Cottrell, F. G.: "Problems in Smoke, Fume, and Dust Abatement," *Smithsonian Annual Report*, **1913**, 653.

positive terminal in the high tension or secondary circuit. (In the original Cottrell plant parallel plates, with wires in between, were used.) Most of the particulate matter in the air under treatment is deposited on the inside surface of the pipe, but slight amounts collect on the wire or negative electrode.

In adapting high tension electric precipitation to the small scale quantitative determination of dust, fume, and smoke in air, Drinker, Thompson, and Fitchet,[13] described a portable apparatus and discussed the theory and history of the principles on which it is based. Their precipitator uses alternating current, and rectifying devices, mechanical or thermionic, are specifically avoided.

Theoretical and Practical Factors in Electric Precipitation.—A particle of dust passing through the tube or pipe of an electric precipitator using either high tension alternating or direct current, is subjected to at least two forces; (1) the force of the

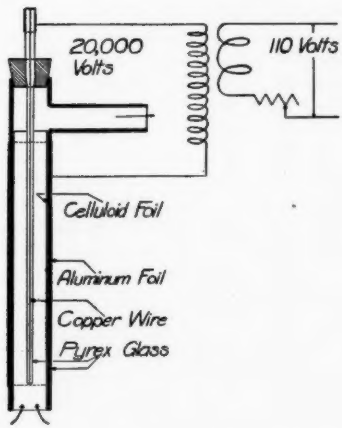


FIG. 8. ELECTRIC PRECIPITATOR SHOWING PRECIPITATOR TUBE AND ELECTRICAL CONNECTIONS

air current carrying the particle along, and (2) the force of electric precipitation tending to drive the particle to the walls of the tube (the collecting electrode). If the force of precipitation is insufficient, the particle passes out of the tube. Theoretically this should be avoidable if the force of precipitation is increased, or the air velocity decreased.

The force causing precipitation is probably due to the formation of charged particles as a result of the corona discharge from the central electrode. From the gas molecules in the tube the corona discharge produces ions, and these ions attach themselves to the particles of dust, fume, or smoke. In the presence of a strong electric field the particles thus charged by ionization are driven towards the collecting electrode where the charges are given up by the deposition of the particles on the surface of the electrode. It is possible to conceive of an insufficient supply or discharge of these ions to care for all the particles in a dense cloud of smoke passing through the precipitator. In such an event, precipitation would be again incomplete and this condition again remediable by increasing the intensity of the

[13] Drinker, P., Thomson, R. M., and Fitchet, S. M.: "Atmospheric Particulate Matter: II The Use of Electric Precipitation for Quantitative Determinations and Microscopy," *Journal of Industrial Hygiene*, **5**, 19, 62 (1923–24).

precipitating force or decreasing the air velocity. Therefore, there should be three distinct factors controlling the efficiency of precipitation:

1. Air velocity
2. Intensity of precipitating force
3. Density of the cloud.

In practical operation of small precipitators, such as that under discussion, it is a simple matter to control at least two of these factors at all times and thus to insure precipitation at close to 100 per cent efficiency.

Other factors of less importance are: (4) diameter, (5) length, (6) dielectric strength of the electrodes. These last three are governed entirely by the capacity of the transformer and can be considered as constant quantities.

Constructional Details of Precipitator. In Fig. 8 are shown the construction of the precipitator tube and the electrical connections. The rheostat in the primary circuit controls the intensity of the field in the secondary or high tension side of the transformer. The high tension leads are well insulated and as short as possible in order to avoid surface leakage. The switch controlling the current is placed on the low tension side. No grounds are needed, nor is any other spark gap than that built on the transformer required.

As a source of air suction a 110 volt universal sewing machine motor is directly connected to a small fan, and air velocity controlled by screw clamps on the rubber connections. A calibrated glass flow meter, placed between the exhaust side of the precipitator tube and suction fan, is used to measure the volume of air per minute drawn through the precipitator tube.

Collecting Electrode. The precipitator tube in which the sample is collected is made of pyrex glass. This is more durable than soft glass and less apt to be punctured by disruptive discharges through the glass tube.

The tube is wrapped tightly with a suitable conducting foil such as aluminium. Inside the tube is a thin transparent celluloid foil which serves as a surface for precipitation. At the completion of a determination the celluloid foil is withdrawn from the tube and the precipitate examined under the microscope, or washed off and weighed, analyzed, or the particles counted.

Precipitating Electrode. The central, or precipitating electrode, is made of fine copper wire placed within a thin-walled, narrow pyrex tube sealed at the bottom, but open at the top. The purpose of the glass tube is to hold the wire in approximate central alignment, and to protect it against corrosion. Wires thick and stiff enough to remain in central alignment without using the glass tube are not so satisfactory. The arrangement shown in Fig. 8 is a slight modification over that described by Drinker, Thompson, and Fitchet,[12] in their original paper, and has proven especially satisfactory in taking samples for microscopic examination. Enclosing the precipitating electrode in a glass tube seems to reduce vibrations and gives more uniform precipitates, but slightly reduces the capacity or precipitating power of the apparatus.

Transformer. The capacity and cost of the electric precipitator depend chiefly on the transformer. If one desires to take samples at fractions of a cubic foot per minute, small and inexpensive 5000 or 10,000 volt transformers weighing about 20 to 30 lb. are satisfactory. The 20,000 volt transformer indicated in Fig. 8 allows rates as high as 2 cu. ft. per min. for most dusts, but weighs about 50 lb. This particular transformer was made by special design and is unnecessarily expensive and heavy for routine field work.

Overloading of Precipitator. By bearing in mind the three important factors:

1. Air velocity
2. Electrical intensity
3. Density of the cloud

low efficiencies in operating the precipitator can be avoided. The point at which particles are precipitated to the wall of the tube depends on the relative strengths of their air velocity, their mass, and the intensity of the precipitating force. If the

FIG. 9. AMMONIUM CHLORIDE FUME. PRECIPITATOR OVERLOADED. (Courtesy of *Journal of Industrial Hygiene*)

FIG. 10. AMMONIUM CHLORIDE FUME, PRECIPITATION COMPLETE. (Courtesy of *Journal of Industrial Hygiene*)

precipitating force is inadequate, the particles will be drawn out over the full length of the tube, as in Fig. 9, and all of them are not precipitated. When functioning properly the precipitate is confined over a short area as in Fig. 10.

The effect of the density of the cloud, or number of particles per cubic centimeter of air, is brought out strongly by comparing precipitation efficiencies with substances like silica dust, zinc oxide fume, and tobacco smoke. The precipitator, with which most of the work in this laboratory is done, has a capacity of about 45 liters per minute for a special silica dust from which large particles are filtered out by blowing the dust cloud through absorbent cotton. The average size of the silica particles in such filtered clouds is three to five microns.

For zinc oxide fume averaging about 0.4 to 0.6 microns[13] the capacity is not over 15 liters per minute, while for tobacco smoke, with particles of about 0.2 to 0.3 microns[14] the capacity does not exceed 10 liters per minute. These differences in capacity of the precipitator are probably due to the fact that a unit volume of dense tobacco smoke contains a vastly greater number of individual discrete particles than a like volume of dusty air in which the individual particles are larger and more apt to collide. In the precipitator it is probable that particles, which have collided and stuck together, behave as a single particle or mass equal to the sum of the individuals.

The Capacity and Precipitating Efficiency of Electric Precipitators.—A convenient method of calibrating apparatus for determining dust, fume, and smoke, has been developed by the Bureau of Mines and fully described by Katz, Longfellow and Fieldner.[15] It consists in passing a steam of air from the dust, fume, or smoke determinator in question, along a beam of light, and the dust stream is then illuminated in approximate proportion to the total surface area of the particles present. This first stream is then matched by a second, of which the density can be accurately controlled. For laboratory work this method is especially useful because it is sensitive and very rapid.

It was estimated by the Bureau of Mines that the Tyndall beam method allowed estimations of 98 per cent efficiency, but that greater efficiencies could not be determined with certainty. In using electric precipitators, the air velocity and intensity of the field can be adjusted so that no smoke or dust stream is visible in the Tyndall box. Consequently the Bureau of Mines method may be simplified and rendered a little more sensitive by using a powerful carbon arc lamp as the source of light, and allowing a convenient fraction of the air escaping from the precipitator to cross the beam from the arc lamp. When no particles are seen by the light beam, it is reasonable to assume that the efficiency of precipitation is greater than 98 per cent. All of the work of calibration can be carried out with efficiencies in this region of 98 to 100 per cent and determinations in which any particles are visible in the beam of light can be rejected.

While this method of judging efficiencies is useful in the laboratory, it is impractical in the field, and leaves one in doubt as to whether precipitation has been complete. A more convenient method, applicable in the field, is to observe the upper limits of precipitation of the celluloid foil. If the precipitator has been over-loaded, Fig. 9, a portion of the precipitate is plainly visible at the top of the foil. When no precipitate is visible on the upper portions of the foil, Fig. 10, it seems safe to conclude that precipitation has been complete. During a determination the same ends can be accomplished by standing near the precipitator and watching for the growing appearance of precipitated particles at the upper end of the aluminium foil. If the operator is willing to make a short experimental run of one or two minutes, a safe air speed and requisite field intensity can be determined without difficulty, and subsequent determinations can be made at efficiencies which are probably very close to 100 per cent.

It seems probable that there is a lower limit to the size of particles which can be caught by electric precipitation. If this is the case, efficiencies appreciably less

---

[13] Green, Henry: "A Photomicrographic Method for the Determination of Particle Size of Paint and Rubber Pigments," *Journal of the Franklin Institute*, **1921**, Nov., 637.

[14] Wells, P. V., and Gerke, R. H.: "An Oscillation Method for Measuring the Size of Ultramicroscopic Particles," *Journal Am. Chem. Soc.*, **41**, 312 (1919).

[15] Katz, S. H., Longfellow, E. S., and Fieldner, A. C.: "Efficiency of the Palmer Apparatus for Determining Dust in Air," *Journal of Industrial Hygiene*, **2**, 167 (1920–21). See also Fieldner, A. C., Katz, S. H., and Longfellow, E. S.: "The Sugar-Tube Method of Determining Rock Dust in Air," U. S. Bur. Mines, *Tech. Paper*, **278** (1921).

than 100 per cent should result in attempting to collect a fume or smoke made up of particles approaching this precipitable limit in size.

Microscopic Examination of Precipitate. The microscopy of the collected sample is usually the most important part of the work in dust problems involving health. The celluloid foil serves as a convenient medium on which to examine the precipitate and offers the great advantage of making it unnecessary to wet the particles with water, an operation which materially changes the physical state of most particles, and may give an untrue picture of the way they occurred in the air prior to collection.

A strip from the celluloid foil is cut with a pair of scissors, allowing sufficient margin to hold the ends of the strip without rubbing off the precipitate on the part to be examined. With the dust side up, the strip is fastened on a microscope slide on which a drop of amyl acetate has been placed. This reagent wets the glass in an even film and is a good solvent for celluloid. The strip of foil sticks fast to the glass and the dust precipitate then lies in a plane slightly above, but parallel to the glass. To free the celluloid from wrinkles, which are visible under the microscope, and sometimes distort the vision, the slide is warmed for a few moments at a temperature of about 70 deg. cent. The preparation can then be examined dry under low and high power magnifications. If the dust side (the upper or exposed side) of the preparation is held for a moment in the fumes of an ether-alcohol bath, the dust becomes fixed to the celluloid, and the preparation made permanent. It can then be examined under oil immersion.

Good mounts can be made of many dust, and some fume precipitates, by placing the dust side down against the glass. If one end of the celluloid strip is fixed on the glass slide by amyl acetate, and the rest of the strip carefully lowered on to the slide without rubbing, the dust precipitate is not disturbed.

Weighing Precipitates. When the weight of the collected precipitate is desired, aluminium foil may be used instead of celluloid, as was recently suggested by T. W. A. Welsh.[16] The advantage in the use of aluminium lies in the fact that it can be brought to the required temperature for drying precipitates, and celluloid cannot, without being affected. The same aluminium foil is satisfactory for the inside as well as the outside of the tube, and should be about 0.0015 in. thick. While we tried Welsh's method on zinc oxide fume only, there seems to be no reason why it is not satisfactory for other substances. The only objection to the use of aluminium instead of celluloid foil, is that celluloid is transparent and the precipitate can be plainly seen. Aluminium foil, being opaque, renders the detection of slight precipitates rather difficult. Welsh's method is more rapid and convenient than that originally suggested by Drinker, Thompson, and Fitchet[12] in which celluloid foil was used and the precipitate washed from the celluloid foil and precipitating electrode, the water evaporated to dryness, and the residue weighed. In either case, whether aluminium or celluloid is used, there is always a slight precipitate on the collecting electrode, and this must be added to the precipitate on the foil.

Transporting Samples. It is desirable to confine field work to the taking of samples and noting general conditions. Work requiring cleanliness of surroundings or of the air, is often impractical. A decided and very useful feature of electric precipitation lies in the fact that precipitation occurs with such violence that samples stick to the foils and will not come off unless the foil is knocked violently.

[16]Union Carbide & Carbon Research Laboratories, Inc., (Personal communication to the writers). Also See Tolman, R. C., *et al:* "An Electrical Precipitator for Analyzing Smokes," *Journal Am. Chem. Soc.*, **41**, 587 (1919).

Consequently the precipitator tubes can be stoppered and mailed or carried to the laboratory for examination.

Because tubes without side arms are cheaper and pack in less space than the tubes shown in Fig. 8, it is sometimes convenient, in field work, to withdraw the foil and precipitating electrode from the precipitator tubes and place them in the

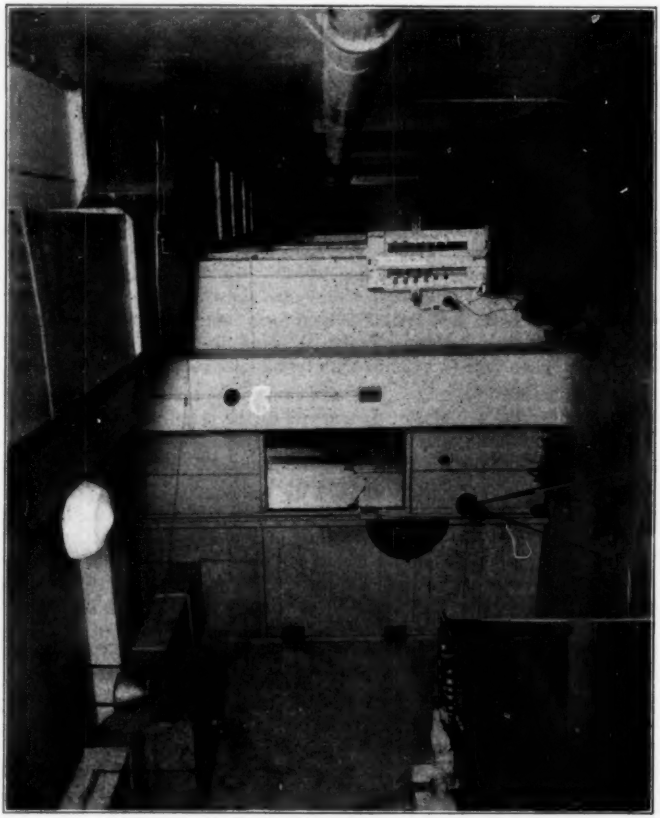


FIG. 11. GAS CABINET—EXTERIOR OF GAS CHAMBER, SHOWING WATER SEAL AND CHAMBER VENT AT UPPER LEFT HAND CORNER, MACBETH ILLUMINOMETER, ELECTRIC PRECIPITATOR AND FLOW-METER. (Courtesy of *Journal of Industrial Hygiene*)

cheaper tubes in which they may then be transported. The precipitator tube can then be used again for taking a new sample.

The Testing of Dust Respirators by Means of the Electric Precipitator

In the plant in which the Owens' samples in Figs. 3–6 were taken, the usual type of "pig snout" dust respirator was in general use by the men. One of the several problems with which we were confronted was to determine the effect of these respirators in filtering out dust particles which would otherwise be inhaled.

To approximate actual working conditions, the respirators were tested against

the same dust found in the plant. A cloud of dried dust was blown into our special dust cabinet,[17] Figs. 11 and 12, and kept in the air by three oscillating fans. Samples of the dusty air were drawn through the filter pad, and the escaping dust caught by an electric precipitator. At the same time a like quantity of unfiltered air was drawn from the cabinet and the dust caught on a second, and precisely similar,

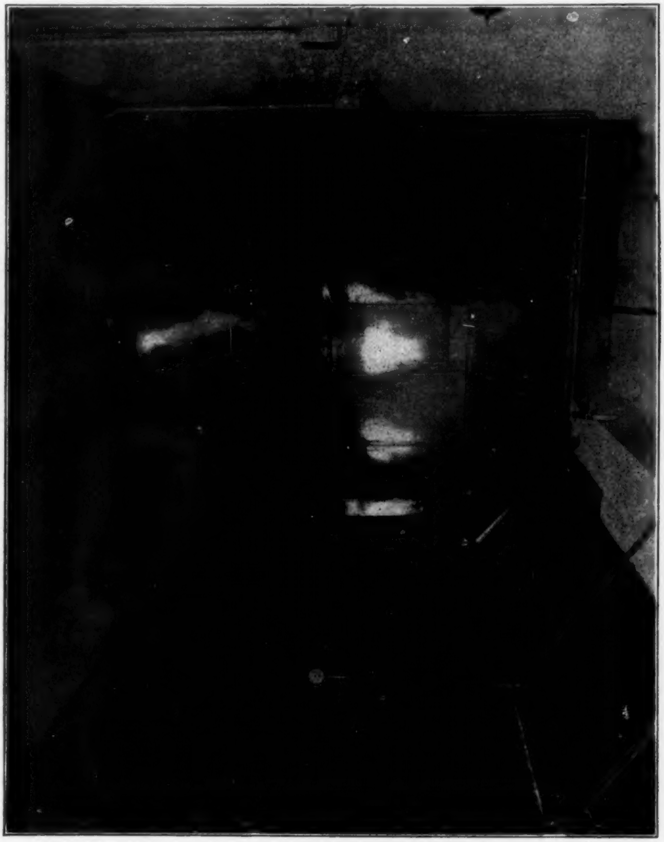


FIG. 12. GAS CABINET—INTERIOR OF GAS CHAMBER, SHOWING ZINC OXIDE FUME FROM ELECTRIC FURNACE AND LAMP USED FOR MEASURING THE LIGHT-OBSCURING POWER OF THE FUME. (Courtesy of *Journal of Industrial Hygiene*)

precipitator. Both precipitators were run for three or more minutes at 25 liters per minute. (A little less than 1 cu. ft. per min.) Inasmuch as it was practically impossible to keep the dust content of the air constant, no attempt was made to approach such a condition. In still air dust settles more rapidly than in air in violent and irregular motion, while high relative humidity, used in several respirator tests, brings in a further factor. By taking samples at determined intervals, the effi-

[17] Drinker, Philip: "Laboratories of Ventilation and Illumination," Harvard School of Public Health, Boston, *Journal of Industrial Hygiene*, **6**, No. 2, 57–66 (1923–24).

ciencies of the respirators under widely different conditions of dustiness could thus be studied.

Twenty-five liters per minute were chosen as representing the approximate maximum rate of inspiration of an average man engaged in moderate steady exercise. Such a hypothetical man would actually breathe a little less than half this amount of air, but his inspirations consume less than half the total breathing time. Therefore his actual inspiration rate is more than double his actual breathing rate. For this reason gas masks, respirators, and equipment of this general description are usually tested at rates of about 1 cu. ft. per min. (28.3 liters). If the subject is undergoing vigorous exercise, his breathing rate may be more than doubled and the tests must be made at correspondingly higher rates of air flow.

By comparing the precipitates from the filtered and unfiltered samples on the celluloid foils, it was possible to determine at a glance the approximate effect of the respirator pad. Fig. 13 shows two foils, one with a precipitate of dust which filtered through the respirator, the other with the dust from the unfiltered air.

FIG. 13. PHOTOGRAPH OF PRECIPITATES FROM RESPIRATOR TESTS. PRECIPITATES CAUGHT ON CELLULOID FOIL, FOIL ON LEFT REPRESENTS DUST IN AIR, FOIL ON RIGHT DUST PASSING THROUGH RESPIRATOR PAD

Since the samples were taken together at similar rates, and for similar periods, they are directly comparable.

After photographing a series of these precipitates, suitable strips were cut off, mounted on slides, and 250 particles on each slide measured by means of a filar micrometer. From these data, the curves in Fig. 14 were plotted. In the filtered air the particles show greater uniformity and are appreciably smaller, as shown by the diameter of the average particle. Practically all particles greater than 5 microns, and by far the greater portion of the small ones, were filtered out by the respirator. Further studies, both chemical and microscopic, showed that the respirators eliminated most of the dust which would ordinarily be inhaled.

In actual use their filtration efficiency probably increases as dust piles up on the upstream side and fills up the voids in the filtering material. But such increased efficiency is acquired at the expense of increased resistance of air flow through the filter, and this is an important consideration to the man wearing the respirator. Furthermore, the fact that these respirators were of undoubted help against the dust in this particular plant, gives no reliable index of their usefulness against a fume like zinc oxide, or a smoke like tobacco which is made of smaller particles and will pass through the respirators much more readily.

Summary

A classification of dusts, fumes, and smokes is suggested and is based on the size of the particles and the methods by which they are liberated into the air. Owens' jet dust counter is described, and its usefulness in general dust problems discussed with the aid of photomicrographs of dusts and fumes collected by his device.

Electric precipitation is discussed in its theoretical and practical aspects as applied to small precipitators for the determination of dusts, fumes, and smokes in air. It is shown that practical operating efficiencies at close to 100 per cent can be

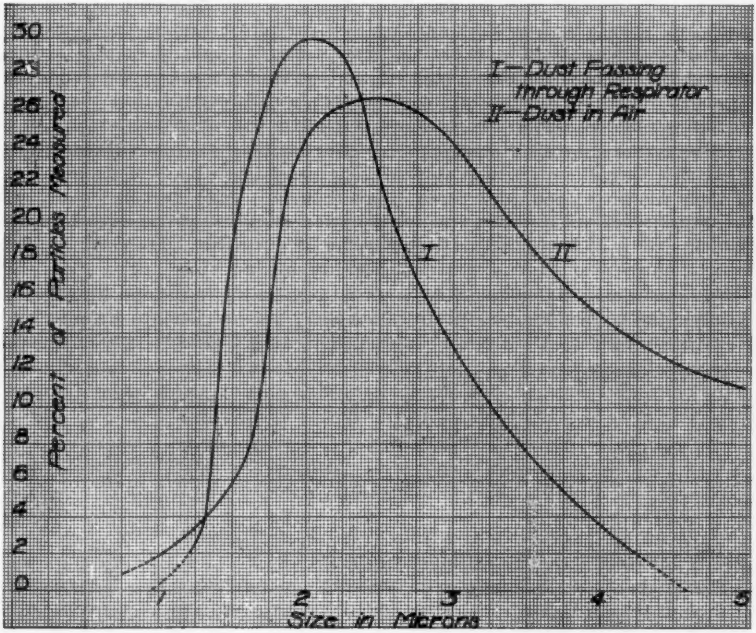


Fig. 14. Size Frequency Curves of Dust in Respirator Tests

obtained in field and laboratory work. Photographs of complete and incomplete precipitation of ammonium chloride fume are given.

A method of mounting transparent celluloid foils on which particles have been precipitated is described in detail. By this method measurements of particle size may be made from the collected sample of dust, fume, or smoke without disturbing the precipitate.

DISCUSSION

President Addams: The Society is very greatly indebted for this fine and valuable paper on this important subject. This paper is open for discussion.

W. H. CARRIER: I would like to ask Dr. Drinker what the advantages of the two methods are, or what was the particular use for which one was used in preference to the other. I would also like to know if any experimental work was done that would show what kind of dusts or fumes or smoke were not collected by the impact method.

PHILIP DRINKER: The difference between the two really gives a distinction which is probably too fussy and impractical for ordinary purposes. If you examine a substance like zinc oxide fume, made by burning zinc in air, you will find particles about 0.6 to 0.7 microns in diameter—much smaller than ordinary dust; or if you use ordinary zinc oxide, as bought in containers, and blow that into the air, instead of getting a dispersed fume, you get particles flocculated together like bunches of grapes—that is under the microscope. If you pass either sample through the Owens' apparatus at high velocity you tend to break up these bunches. Although you will undoubtedly obtain a measure of how much dust there is in the air you do not, necessarily, obtain a measure of the individual particles.

That is really the chief distinction we have between the two. Personally, I think that Owens' apparatus is an extremely useful one. You can see from one of those slides that you get a dense ribbon if there is much dust in the air, but it is very easy to get a ribbon so dense that you can't do anything with it except photograph it. You couldn't possibly count the particles. Obviously Owens' counter is not adapted to use in very dusty atmospheres.

With our electric precipitator, if the air is very dusty, we run it a little faster. Then we can get a slide upon which we can really see individual particles and we feel that we may get them in the state in which they originally were in the air.

W. H. CARRIER: How about smoke?

PHILIP DRINKER: We can catch that just about as easily but can't run it quite so fast. The general efficiency of the precipitator seems to depend on the actual number of particles or aggregates which are in the air. With the smoke you can make a much denser cloud; you can get more particles per cc. than you can with the dust.

W. H. CARRIER: I wanted to find out exactly how Owens compared with the electric precipitator in the matter of smoke as well as fume.

PHILIP DRINKER: Smoke to us means burning something that is carbonaceous like oil or tobacco. That is a rather fine distinction; and though you can make particles of smoke, which are of about the same size as you can make by burning a substance like zinc, when you make tobacco smoke you form water which has a profound effect on the particles. If you take a sample of tobacco smoke with Owens' apparatus, all you get is a wet tarry ribbon; you can't see any individuals no matter how small a sample you take or how thin the smoke is in the air. With our machine, if we deliberately run it fast, we can obtain one layer of particles. However, all we see under the microscope are tiny little droplets colored with tar.

No. 715

THE A-A DUST DETERMINATOR

By MARGARET INGELS,[1] PITTSBURGH, PA.

MEMBER

Part I

THE dust chart presented in the JOURNAL OF THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS February, 1924, was in such a form that reading values from it accurately was difficult and the chart is one that must be used extensively with the A-A Determinator, therefore work was done to simplify the presentation of this information. The entire problem was approached in a different manner than heretofore, as is explained in the following details of the construction of the dust chart.

The New Dust Chart

The first phase of the dust problem which was presented in connection with the A-A Dust Determinator has been the most difficult part of the problem to solve. This phase is to determine the relation between the amount of dust deposited on the filter medium and the increase in resistance across the filter medium, and to determine the effect of the initial resistance on this relation. That is, (1) Does 2 in. increase in the resistance of the filter medium mean twice as much dust as 1 in. increase in resistance? (2) Does a resistance increase of 1 in. mean the same dust deposit when the initial resistance is 5 in. as when the initial resistance is 6 in.?

There is no way of producing known degrees of dustiness and no accepted method of measuring the degree of dustiness for any air. The methods used by various scientists are not as sensitive or as reliable as the A-A Determinator, therefore, this instrument will have to be checked against itself.

The tests, reported in the February, 1924 issue of the JOURNAL are used for checking all theoretical values. The tests were made by measuring the increase in resistance across the filter medium when room air, air containing small coal dust particles and air containing large coal dust particles was sampled. Each test consisted of six simultaneous determinations and a total of seventeen tests was made. The results of these tests are given in Table 1.

It can be seen from this table that the increase in resistance is not a direct measure of the dust deposit. For instance, in test 2, *room air*, the resistance increases

---

[1] Research Head, A.S.H.&V.E. Research Laboratory.

Presented at the Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Boston, January, 1925.

vary from 3.45 to 6.05 in., but also the initial resistances of the papers vary from 4.2 to 6.7 in. The dust deposits are equal, therefore a correction must be made for the variation in initial resistances of filter papers. The initial resistances of the papers were plotted against the final resistances for each test. These curves were plotted on logarithmic paper, and each test was a straight line curve but the slope

TABLE 1. RESULTS OF TESTS WITH AIR CONTAINING LARGE AND SMALL COAL DUST PARTICLES

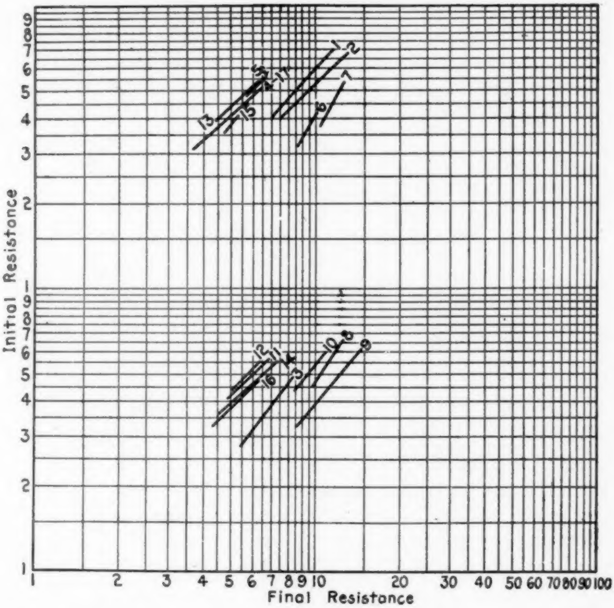
Room Air	1	2	3	4	5	6
1.						
Initial pressure drop	4.6	5.8	5.6	5.7	6.2	4.7
Final pressure drop	7.8	9.9	9.2	9.8	10.4	7.35
2.	6.7	6.2	4.2	4.6	5.7	5.45
	12.75	11.4	7.65	8.75	11.3	10.45
3.	4.15	4.4	3.2	3.35	3.1	3.55
	7.25	8.1	6.0	6.1	6.15	5.6
4.	4.15	4.05	3.25	4.2	5.1	3.5
	5.0	4.85	3.95	5.1	6.4	4.0
5.	5.15	5.2	5.2	5.4	5.0	4.9
	6.1	6.2	6.2	6.35	6.0	5.8
6.	3.5	4.05	4.0	3.6	3.3	3.9
	9.2	10.0	9.9	9.25	8.9	9.8
7.	4.0	4.5	4.7	3.55	4.0	5.15
	11.0	11.8	11.5	9.4	10.0	11.7
8.	4.65	5.6	5.8	6.55	6.05	5.45
	10.3	11.0	11.8	13.0	11.5	11.3
9.	4.8	3.8	4.8	5.1	3.65	4.5
	11.7	10.4	12.05	12.45	9.4	10.8
10.	4.75	5.9	4.6	4.85	5.3	4.7
	9.2	10.8	8.65	9.5	9.8	8.65
Small size coal dust						
1.	4.65	5.15	4.8	4.4	4.9	5.65
	5.65	6.2	5.7	5.2	5.9	6.85
2.	5.0	5.2	4.6	4.8	5.2	5.2
	5.7	6.0	5.25	5.55	6.05	5.9
3.	4.7	4.1	4.85	5.1	4.05	4.3
	5.4	4.65	5.5	5.85	4.5	4.85
Large size coal dust						
1.	4.3	5.3	3.8	3.9	3.9	3.9
	5.4	6.9	4.8	4.95	4.9	5.0
2.	3.7	3.9	3.95	3.9	3.8	4.0
	4.9	5.1	5.2	5.2	5.0	5.1
3.	4.75	4.05	3.65	4.1	3.95	4.5
	6.3	5.35	4.85	5.55	5.05	5.8
4.	3.8	4.35	4.4	4.35	4.05	4.3
	4.65	5.3	5.35	5.3	4.85	5.25

of the lines varied, becoming greater with the degree of dustiness. (See Fig. 1.) The relative degrees of dustiness for the different tests are not known, therefore, the ratio of the slope to dustiness could not be established here. The correction to be made for the various initial resistances cannot be determined here for the correction is a function of the degree of dustiness.

Known relative amounts of dusts must be collected on different filter papers and the effects of known dust compared for papers with various initial resistances. It is impossible to get the relations of the dust deposits, therefore instead of the

area of the papers being blocked up by many minute particles of dust, the papers were blocked off by mica discs of known diameters.

Tests were run on several filter papers and the pressure drop in inches of water was plotted against the volume of air in cubic feet per minute for the papers when their full areas were exposed for air passage and when known per cents of their full



Curves 1–10 Room Air Curves 11–13 Small Coal Dust Particles Curves 14–17 Large Coal Dust Particles

FIG. 1. CHART SHOWING RESULTS OF SERIES OF TESTS

areas were blocked off by the mica discs. The discs and the way they were placed on the filter paper during the tests, are shown in Fig. 2.

Test Data

Table 2 gives the data obtained in these tests, and in Fig. 3 are shown the curves plotted from this data.

It can be seen from these curves that an equal decrease in area causes a greater increase in pressure difference across the paper when the paper has a high initial resistance than when it has a low initial resistance.

The A-A Dust Determinator handles 0.4 cu. ft. per minute, therefore the pressure drops for each paper with its decreases in area are read for this volume of air, and compiled in Table 3.

Fig. 4 shows the curves on logarithmic paper for each filter paper with its per cent

TABLE 2. DATA FOR TESTS OF FILTERS WITH FULL AND PARTIAL AREAS EXPOSED

Filter No.	C. F. M.								
		100%	91.5%	82.6%	74%	65.6%			
	0.158	1.5	1.6	1.7	1.9	2.05			
	0.233	2.35	2.4	2.65	2.9	3.2			
	0.336	3.5	3.7	4.05	4.45	4.9			
1.	0.417	4.3	4.7	5.05	5.6	6.2			
	0.482	5.0	5.45	5.9	6.55	7.25			
	0.54	5.65	6.1	6.65	7.3	8.05			
	0.592	6.2	6.7	7.3	8.1	8.95			
		100%	73.6%						
	0.158	1.45	2.05						
	0.233	2.05	3.0						
	0.336	3.25	4.55						
2.	0.417	4.05	5.7						
	0.482	4.75	6.7						
	0.54	5.3	7.5						
	0.592	5.83	8.3						
		100%	93.6%	87.3%	80.9%	74.5%	68.2%	61.9%	55.6%
	0.158	1.4	1.5	1.5	1.55	1.55	1.65	1.75	1.95
	0.233	2.0	2.1	2.15	2.25	2.35	2.45	2.7	2.9
	0.336	3.0	3.15	3.35	3.45	3.65	3.85	4.1	4.45
3.	0.417	3.7	3.95	4.15	4.3	4.5	4.85	5.1	5.65
	0.482	4.35	4.55	4.85	5.05	5.3	5.65	6.05	6.6
	0.54	4.85	5.1	5.45	5.7	5.9	6.3	6.75	7.45
	0.592	5.3	5.6	5.9	6.15	6.55	6.95	7.5	8.2
		100%	57.6%						
	0.158	1.3	1.95						
	0.233	1.9	2.9						
	0.336	2.85	4.5						
4.	0.417	3.55	5.7						
	0.482	4.1	6.7						
	0.54	4.6	7.5						
	0.592	5.05	8.25						
		100%	73.3%	64.8%	56.2%	47.5%			
	0.158	1.35	1.8	1.9	2.1	2.5			
	0.233	2.0	2.75	2.95	3.4	3.9			
	0.336	3.05	4.2	4.6	5.25	6.1			
5.	0.417	3.8	5.3	5.8	6.6	7.7			
	0.482	4.45	6.2	6.85	7.8	9.1			
	0.54	4.95	7.0	7.7	8.7	10.2			
	0.592	5.45	7.7	8.5	9.7	11.35			
		100%	93.6%	87.3%	70.8%	64.5%	37.8%		
	0.29	2.9	3.0	3.1	3.6	3.9	6.6		
6.	0.336	3.45	3.6	3.75	4.25	4.6	7.9		
	0.378	3.9	4.05	4.2	4.8	5.2	8.95		
	0.417	4.3	4.5	4.65	5.3	5.75	10.1		
		100%	91.5%	82.6%	40.4%	27.7%			
	0.29	3.0	3.2	3.45	6.5	8.3			
7.	0.336	3.55	3.8	4.05	7.8	9.8			
	0.378	4.0	4.25	4.5	8.8	11.1			
	0.417	4.4	4.75	5.75	9.8	12.4			

of area plotted against the pressure drop across it when the volume of air through the paper is 0.4 cu. ft. per minute. The average slope of the lines plotted in this way is —1.18. Therefore the relation of the per cent area to the pressure drop can

be set up by the equation:

$$\text{Area} = c\,P.\,D.^{-1.18} \qquad (1)$$

With this equation the figures in Table 4 were computed.

The last column in Table 4 was computed by assuming the area at 2 in. pressure drop to be equal to 100 per cent which means c in Equation 1 is 226.76. This is an arbitrary value but the absolute area is not obtainable and is not necessary for this work.

In Fig. 5 is plotted the per cent area against the pressure drop as computed in Table 4. As the relative areas are equivalent to the per cent areas when compared to the same 100 per cent, the per cent sign will be dropped in the future readings.

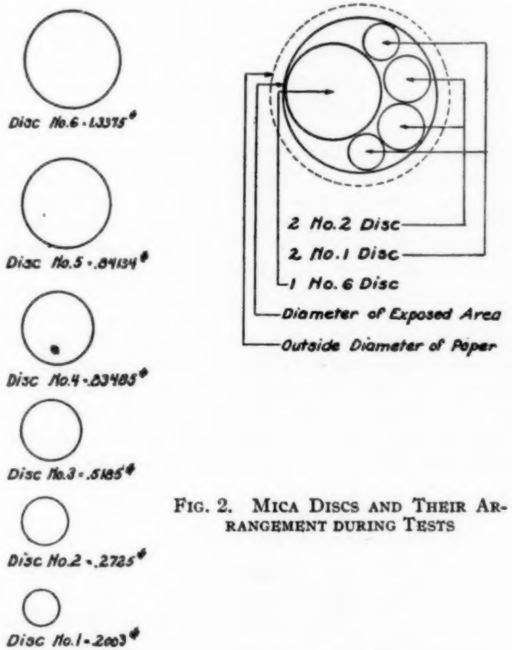


FIG. 2. MICA DISCS AND THEIR ARRANGEMENT DURING TESTS

TABLE 3. PRESSURE DROPS FOR VARIOUS FILTER PAPERS FOR AIR VOLUME OF 0.4 CU. FT. PER MINUTE

Paper No. 1		No. 2		No. 3		No. 4		No. 5		No. 6		No. 7	
P. D.	% A	P. D.	% A	P. D.	% A	P. D.	% A	P. D.	% A	P. D.	% A	P. D.	% A
4.15	100	3.88	100	3.58	100	3.42	100	3.65	100	4.12	100	4.26	100
4.45	91.5			3.78	93.6					4.28	93.6	4.57	91.5
4.86	82.6			4.0	87.3					4.5	87.3	4.86	82.6
5.36	74.0	5.5	73.6	4.15	80.9					5.13	70.8		
5.94	65.6			4.35	74.5			5.08	73.3	5.53	64.5		
				4.62	68.2			5 6	64.8				
				4.95	61.9			6.36	56.2			9.42	40.4
				5.41	55.6	5.41	57.6	7.36	47.5	9.65	37.8	11.81	27.7

TABLE 4. RELATION OF PER CENT AREA TO PRESSURE DROP

No. P. D.	Log. No.	Log x 1.18	No.	$\frac{1}{\text{P. D.}^{1.18}}$	% A
2	0.30103	0.3552	2.266	0.441	100
3	0.47712	0.5620	3.648	0.274	62
4	0.60206	0.71043	5.133	0.195	44.2
5	0.69897	0.872478	6.680	0.1495	33.9
6	0.77815	0.91822	8.284	0.121	27.4
7	0.8451	0.99722	9.940	0.1006	22.8
8	0.90309	1.0656	11.63	0.086	19.5
9	0.95424	1.1259	13.36	0.075	17.0
10	1.00000	1.1800	15.14	0.0656	14.9
11	1.04139	1.2288	16.94	0.059	13.4
12	1.0792	1.2734	18.77	0.0532	12.07
13	1.1139	1.3143	20.62	0.0484	11.0
14	1.14613	1.3524	22.51	0.0451	10.23
15	1.1761	1.3878	24.4	0.041	9.3

The data in Table 1 are used to check the curve shown in Fig. 5. The areas for each initial resistance and for each final resistance are read from this curve and the difference in areas computed for each dust deposit. Table 5 gives the decrease in area and also the initial resistance of the filter paper. The table shows that the curve does not fully correct for the variation in the initial resistance. The theoretical curve does not give a true correction for the test data. This is probably due to the mica disc not affecting the filter medium in the same way that dust deposits would affect it. The mica discs block off large areas at one time, leaving the exposed area unchanged. The dust particles, when sufficient to block off the same area are evenly distributed and the entire paper is changed. A further correction must be made for this difference in effect.

The values for the decreases in area in Table 5 were plotted on logarithmic paper against the initial resistance and are shown in Fig. 6. The average slope of these lines is —1.26, therefore from these curves the following relation is set-up.

$$A_1 - A_2 = K \frac{1}{P_1^{1.26}} \qquad (2)$$

A_1 = Initial area
A_2 = Final area
P_1 = Initial pressure difference in inches of water

It is evident that for each degree of dustiness, the difference in area is a function of the initial resistance but the decrease in area for various degrees of dustiness is a function of the dust deposit and of the initial resistance of the paper.

For each degree of dustiness the relation of the decrease in area to the initial resistance of the paper is:

$$A_1 - A_2 \frac{1}{P_1^{1.26}}$$

That is

$$A_1 - A_2 = X \frac{1}{P_1^{1.26}}$$

TABLE 5.

Room Air	1	2	3	4	5	6
1.						
Initial pressure drop	4.6	5.8	5.6	5.7	6.2	4.7
Decrease in area	17.7	13.7	13.8	14.0	12.2	15.2
2.	6.7	6.2	4.2	4.6	5.7	5.45
	12.9	13.6	21.4	20.2	16.4	16.8
3.	4.15	4.4	3.2	3.35	3.1	3.55
	20.6	20.6	30.3	27.4	34.2	21.0
4.	4.15	4.05	3.25	4.2	5.1	3.5
	8.7	8.3	11.5	8.7	7.8	7.5
5.	5.15	5.2	5.2	5.4	5.0	4.1
	5.8	6.0	6.0	5.4	6.4	6.2
6.	3.5	4.05	4.0	3.6	3.3	3.9
	35.2	28.7	29.4	33.8	38.0	30.0
7.	4.0	4.5	4.7	3.55	4.0	5.15
	31.0	26.3	23.0	34.7	29.5	20.7
8.	4.65	5.6	5.8	6.55	6.05	5.45
	22.6	16.5	16.3	13.5	14.5	18.0
9.	4.8	3.8	4.8	5.1	3.65	4.5
	23.2	32.6	23.6	21.6	33.2	25.5
10.	4.75	5.9	4.6	4.85	5.3	4.7
	19.4	14.6	20.0	19.1	16.5	19.2
70 Micron dust						
1.	4.65	5.15	4.8	4.4	4.9	5.65
	7.5	6.4	6.6	7.45	6.7	6.0
2.	5.0	5.2	4.6	4.8	5.2	5.2
	4.5	5.0	5.5	5.7	5.1	4.6
3.	4.7	4.1	4.85	5.1	4.05	4.3
	5.5	6.2	4.5	4.7	5.0	5.3
140 Micron dust						
1.	4.3	5.3	3.8	3.9	3.9	3.9
	8.6	8.5	11.2	11.0	10.5	11.5
2.	3.7	3.9	3.95	3.9	3.8	4.0
	13.6	12.3	12.4	12.9	13.0	11.2
3.	4.75	4.05	3.65	4.1	3.95	4.5
	10.1	12.1	14.0	11.7	11.5	10.2
4.	3.8	4.35	4.4	4.35	4.05	4.3
	9.8	8.2	8.3	8.2	8.4	8.4

TABLE 6.

Kind of Screen	Velocity	Concentration of Dusty Air	Approximate % Efficiency
130 mesh	5000	Coal Dust 9.0 A.A. Units	30
200 mesh	700	Coal Dust 6.0 to 18.0 Units	25
350 x 20 mesh	700	Coal Dust 8.0 to 25.0 Units	30–50
325 mesh	700	Coal Dust 13.0 Units	60
Linen	46.0	Coal Dust 13.0 Units	65
Fume cloth	46.0	Coal Dust 7.0 Units	65
Fume cloth	46.0	Room Air 16.0 Units	20

NOTE: The efficiency varies for different tests depending on how well the dust from the previous tests are removed from the medium.

Approximate efficiencies are given because the actual efficiency varies, and becomes greater as the test is continued.

where X is the degree of dustiness of the air handled. From Equation 1 and 2 may be written the following equation:

$$\frac{226.76}{P_1^{1.18}} - \frac{226\cdot 76}{P_1^{1.18}} = X\frac{1}{P^{1.28}}$$

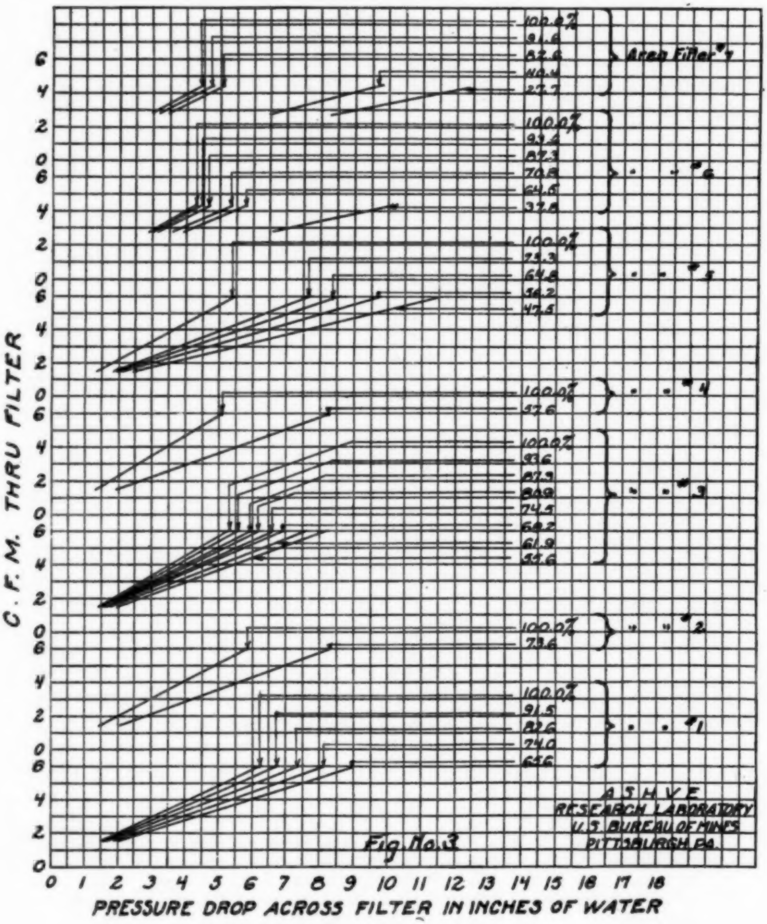


FIG. 3. CURVES PLOTTED FROM TEST DATA RECORDED IN TABLE 2

Therefore

$$X = 226.76 \left(P_1^{0.08} - \frac{P_1^{1.26}}{P_2^{1.18}} \right) \quad (3)$$

Values for X were computed from Equation 3 for each initial resistance and each final resistance for every inch from 2 to 15 in. The values for X thus obtained were plotted against the final resistances for each initial resistance, and are shown in Fig. 7.

Values were read from the curves in Fig. 7 and replotted in Fig. 8. The relations

of degree of dustiness to the initial and final resistance is shown by straight lines with this arrangement. This set of curves is the revised dust chart.

The degree of dustiness is represented by an arbitrary scale, and in the second part of this report will be given equivalents in weights of some known dusts.

Part II

Calibration by Weight of the A-A Dust Determinator

The principal criticism of the A-A Dust Determinator is that it takes into account very small particles of dust. At present there is no dust remover on the

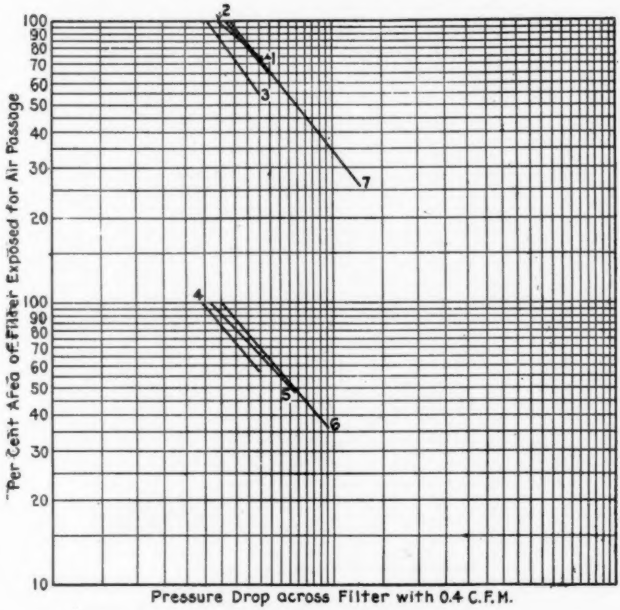


FIG. 4. CURVES FOR EACH FILTER PAPER WITH AIR VOLUME OF 0.4 CU. FT. PER MINUTE

market which has a high cleaning efficiency on air containing small dust particles.

Most commercial air cleaners will remove only large dust particles, and because the future of the A-A Dust Determinator depends on the ventilating engineer, plans were made to change the determinator so it would act more in accordance with the commercial apparatus it was designed to test.

The first and only attempt to change the determinator was by the use of various materials for filter medium instead of the very efficient chemical filter paper. To get necessary information, the dust removal efficiencies of the materials were tested against the A-A Dust Determinator as originally designed.

The drawing in Fig. 9 shows the laboratory set-up for obtaining the efficiencies

of the various mediums. *A*, in the drawing, is an enclosed laboratory hood with a volume of approximately 24 cu. ft. The dust liberator, developed in this Laboratory, and described in the February, 1924 JOURNAL, was placed in the hood, and used to put a known dust into the air when the room air was practically clean. The air drawn through the testing apparatus was taken from the hood.

The medium to be tested was put in frame No. 2. The gross area for air passage of the filter frame has a diameter of 2 in. When wire mesh screens were tested,

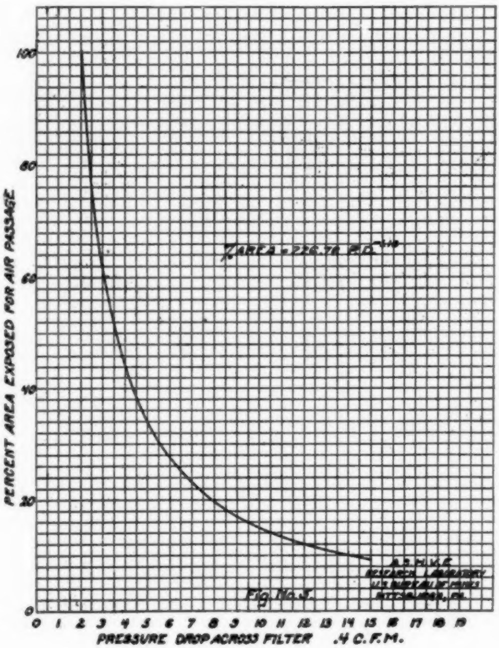


FIG. 5. CURVE SHOWING PER CENT AREA PLOTTED AGAINST PRESSURE DROP AS COMPUTED IN TABLE 4

the exposed areas were necessarily much smaller because the net areas of the screens were great in comparison to the gross areas and the velocities through the screens must be made sufficient to give readable pressure drops across the screens. The screens were soldered on to copper discs covering orifices made the diameter to give the desired exposed area. The velocities through the screens were varied by two means, one by changing the quantity of air drawn through the screens for different tests, the other by using different sizes of orifices covered by the screens. Tests were made on screens of various mesh.

Air was drawn from the hood as shown in the drawing and divided, part being drawn through apparatus 1, the other part being drawn through apparatus 2 and on through apparatus 3 in series with apparatus 2.

The dust concentration of the incoming air was determined by apparatus 1. The dust concentration of the air leaving the trial medium was determined by apparatus 3. Instruments 1 and 3 are A-A Dust Determinators as originally designed handling 0.4 cu. ft. per minute through A. D. Little chemical filter paper through an exposed area having two inches diameter.

The flow meter *J* in the drawing was used when air quantities greater than 0.4 cu. ft. per minute were drawn through the trial medium to make up the additional air above 0.4 cu. ft. per minute.

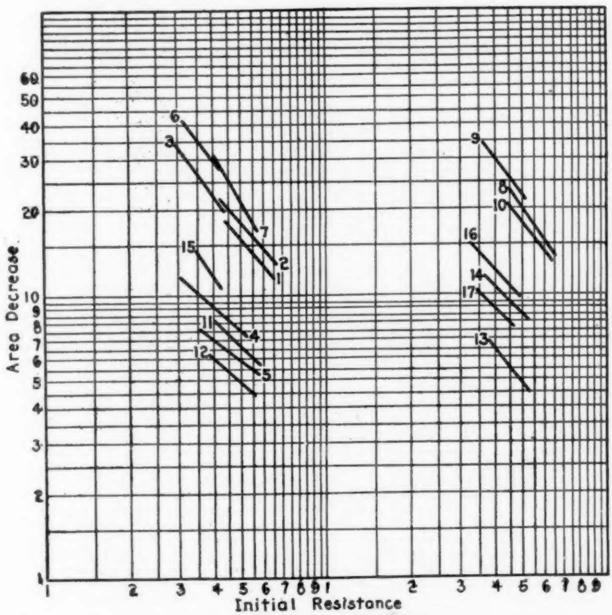


FIG. 6. CURVES SHOWING VALUES FOR DECREASES IN AREA IN TABLE 5 PLOTTED AGAINST INITIAL RESISTANCE

The data taken for tests made with this arrangement were: The concentration of dust entering the trial filter medium, the amount of air passing through the trial medium, and the specifications of the medium. The dust concentration measurements were made according to the dust chart given in Part I of this report. Table 6 gives the information found from these tests.

The entire scheme of introducing a new medium for the A-A Dust Determinator in order to obtain a medium that would not be affected by small dust particles, was not attractive to the investigator, so a different method of obtaining the approval of the ventilating engineer was studied.

It has always been the desire of those responsible for the A-A Dust Determinator to get a relation between the resistance increase and the weight of the dust deposit. It is desirable to get this relation with several known dusts whose specific gravities

can be determined. The value of a quantity of dust measured by weight has no significance if the specific gravity of the dust is not known.

The Research Laboratory was fortunate in obtaining a room at the Bureau of Mines where the dust experiments could be carried on exclusively. Accurate and sensitive balances were in the same room where the tests were made so that weights

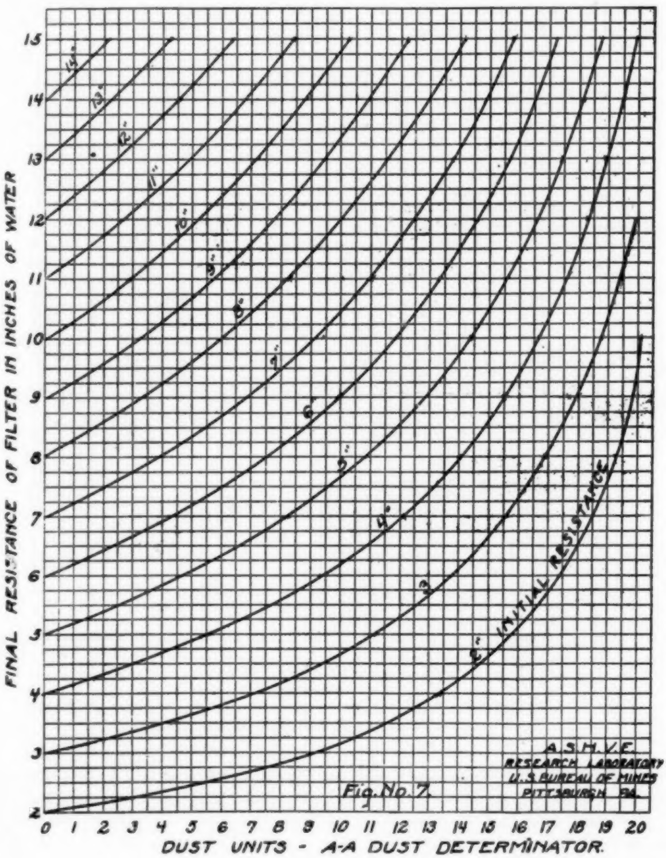


FIG. 7. VALUES FOR X IN EQUATION 3

could be made of the filtering medium immediately before and after tests, and under the same air conditions that prevailed during the tests. With this arrangement for work, tests were again made to get the weight-resistance relations.

The Laboratory set-up shown in Fig. 9 was used for these tests. Instruments 1, 2 and 3 were used as A-A Dust Determinators, that is, each had A. D. Little

chemical filter paper as a medium, and air was drawn through them at the rate of 0.4 cu. ft. per minute. Tests were made as follows: Three filter papers were each weighed in the open, the weight being made to the fifth decimal place in grams. The papers were put into their frames and air drawn through them for one hour. The increase in resistance across each paper was recorded every five

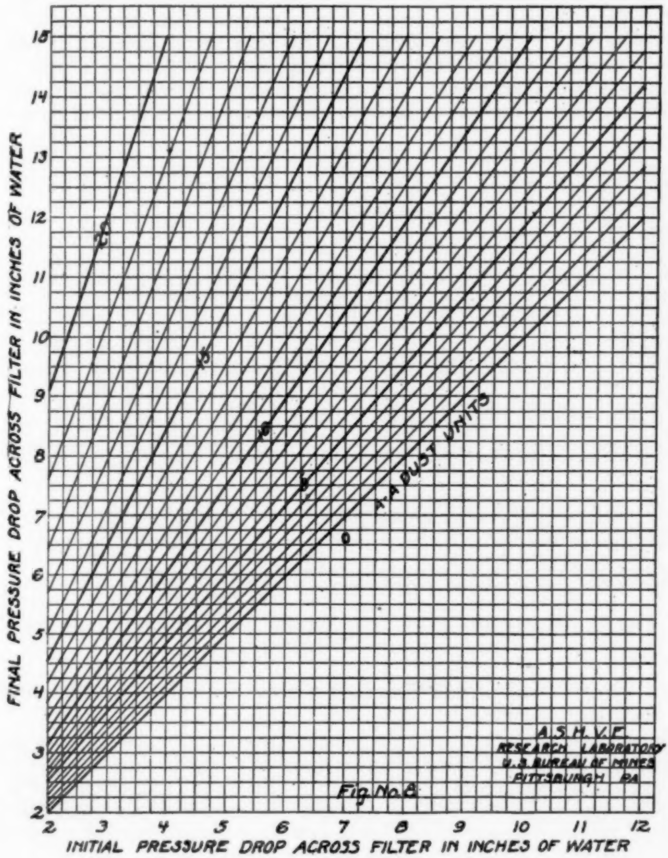
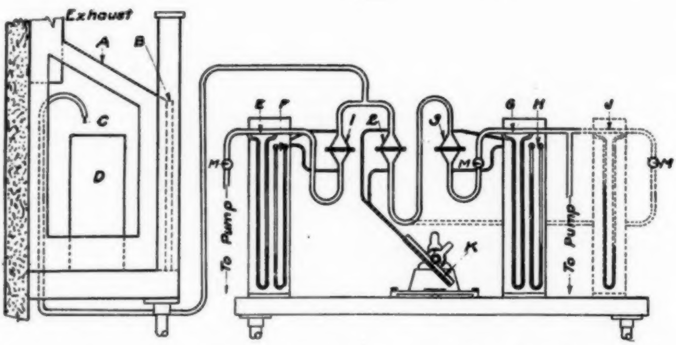


FIG. 8. REVISED DUST CHART

minutes. At the end of one hour the papers were removed, and again weighed. The deposit on papers in instruments 1 and 2 should be approximately the same. The deposit on the paper in instrument 3 should be zero, as the air containing the dust had passed through instrument 2 and the medium is very efficient. Assuming that no dust was deposited on the paper in instrument 3 any change in weight of its medium was due to moisture. It was found that often the paper in this instru-

ment lost weight, or gained weight. By correcting the difference in weights of the other two papers according to this change in weight due to the change in moisture content, the actual weight of the dust deposit was obtained. The dust deposit was also measured by the resistance increase, the values figured in A-A dust units by means of the dust chart given in Part I of this report.

Points were plotted using the weight of the dust deposit as the ordinate and the A-A Dust Units as the abscissae. The experimental data are shown in Fig. 10. The points are very scattered both when the dust deposited was that found in ordinary room air, and when it was that of coal dust added to clean room air. The weights were exceedingly small and a possible error of 0.0001 grams is probable in each weighing. The points show the characteristics of the curves for the two



1 A-A Dust Determinator	C Sampling Tube	H Pressure Difference Gage
2 A-A Dust Determinator	D Dust Liberator	J Flow Meter
3 A-A Dust Determinator	E Flow Meter	K Inclined Manometer Pressure Difference Gage
A Inclosed Hood	F Pressure Difference Gage	M Control Valves
B Sliding Door	G Flow Meter	

FIG. 9. LABORATORY SET-UP FOR WEIGHT CALIBRATION OF THE A-A DUST DETERMINATOR

dusts. The test data of resistance increase may be converted to the weight of dust by the use of these curves.

The curves show an interesting relation of the natural room air dusts and the artificial coal dust. The difference of the two curves may be caused by two things. First, the specific gravity of the dusts may not be equal. This would tend to give a greater slope to the curve of the dust with the greater specific gravity but the shape of the curve should not change to any great extent. Second, the dust put into the air artificially must necessarily be of larger particles than that found in the natural air. It seems that the smaller the dust particles the greater is the increase in resistance for the same weight. This may be caused by the small particles imbedding themselves in the filter paper while the large particles stack up on the surface of the paper. If it is desired to convert from resistance increase caused by a dust deposit to the weight of the dust deposit, the A-A Determinator should be calibrated for all principal dusts.

The characteristics shown by the weight-resistance curves (the resistance having been corrected by the dust chart and shown in dust units) prove the theory that

was thought to be true by some engineers, namely, that small weights of dust showed a greater effect on the resistance basis than large weights of dusts. To illustrate this, refer to the curves in Fig. 10. Comparing the weights of 4 A-A dust units and 12 A-A units the weights for room air are 0.00007 and 0.00034 and for coal dust are 0.00023 and 0.0016, respectively. In neither case are the weights a 3 to 1 ratio although this is the ratio of the dust units measured with the dust determinator.

It is evident that the measure of resistance increase is not a function of the weight of dust alone, but a function of the size of the dust particle also.

The fact that the resistance increase varies with the size of the dust, and the specific gravity of the dust, does not lessen the value of the A-A Dust Determinator.

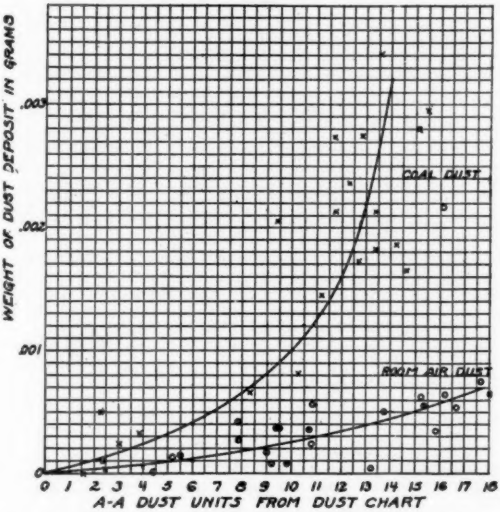
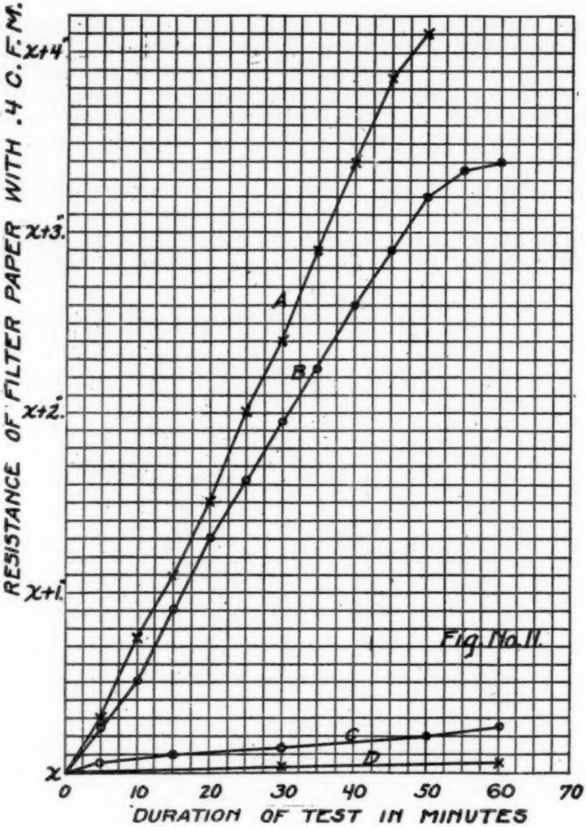


FIG. 10. CURVES SHOWING EFFECT OF WEIGHT OF DUST ON RESISTANCE

It can be seen that straight lines drawn through the curves in Fig. 10 will not part from the curves in any appreciable amount up to 10 A-A dust units. Ten A-A units of dusts means an increase in resistance from 2 to 3.1 in. or from 9 in. to 13.4 in. Usually tests run about 4 in. initial resistance which means the resistance can build up to 6.1 in. and keep within the working limits of the dust determinator. Any test can be limited to this increase in resistance. If the air is extremely dusty, the test can be shortened.

The greatest concentration of dust found in room air during these tests gave an increase in resistance of 2 in. in 25 min. Coal dust gave an increase in resistance of 2 in. in 30 min. when the heaviest concentration of it was used. It was only over very long periods of time that the large deposits were obtained for the weights in these tests.

When determining the dust content of any air the resistance of the paper read



X = Initial Resistance of Filter Paper
A = Heaviest Concentration, Room Air Dust
B = Heaviest Concentration, Coal Dust
C = Lightest Concentration, Coal Dust
D = Lightest Concentration, Room Air Dust

FIG. 11. CURVES SHOW METHOD OF PLOTTING TEST DATA AND ILLUSTRATE THE PROCESS OF DUST CONCENTRATIONS WITH THE A-A APPARATUS

at equal time intervals should be plotted against the time while the test is in progress. As long as the resistance builds up in a straight line, the test can be continued but as soon as the curve starts to turn towards a horizontal line the test should be stopped, a new filter paper put in the frame, and another test started.

There are two factors which may make the curve turn. One is a decrease in the dust content of the air, and to make sure that this is what might be happening two tests at least should be made in immediate succession. The second test would show if there is a dropping-off of dust. The other factor is due to the nature

of the dust deposits. If the paper is "saturated" with dust, that is the dust no longer imbeds itself in the paper but stacks up on top of the paper, the resistance does not increase as rapidly. When the paper reaches this stage, a new paper should be put in the frame because the rate of resistance increase is no longer a measure of dust. It was found during the tests made that the papers reach a state of saturation at a lower pressure difference for large dust particles than for small dust particles.

The curves in Fig. 11 show the method of plotting the data as it is taken in the test, and illustrate completely the process of determining dust concentrations with the A-A Dust Determinator. The resistance of the filter paper is read at the beginning of the test and at 5 min. intervals, throughout the test and the resistance is plotted against the time of reading. As long as this graph remains a straight line, the test may be continued. The length of the test is arbitrary, otherwise. Suppose a test is run for 30 min. on air with a dust concentration equivalent to that for curve *A* in Fig. 11. The resistance builds from x to x + 2.5 in. in 30 min. Referring to the dust chart the number of A-A Dust units is read for this resistance increase. Dividing the number of dust units by (0.4 cu. ft. per minute x 30 min.) gives the units per cubic foot. Now if it is desirable to know the weight of dust per cubic foot curves in Fig. 10 are used. The total A-A Dust units found by the test are read in terms of weight for the specific dust tested. This weight is divided by (0.4 cu. ft. per minute x 30 min.) for the weight per cubic foot of air sampled. It is evident that the resistance-weight relation should be estimated for various kinds of dust. At present it is sufficient to use the room air dust curve for naturally dusty air, and the coal dust curve for air containing artificial dusts as that found in factories, and so forth, until provisions are made for calibrating the instrument for other known dusts.

No. 716

FURTHER DATA ON INFILTRATION OF AIR THROUGH BUILDING OPENINGS

By C. C. Schrader, Pittsburgh, Pa.

MEMBER

SOME interesting results of tests made on door and double casement windows are reported in this article and amplify the data given in two previous articles[1] covering the subject of leakage around double hung windows, plain and weatherstripped.

Nonweatherstripped Windows and Doors

Casement windows and doors differ from double hung windows in that their position is fixed in the opening, whereas double hung windows are free to move from side to side, or backward and forward between the stops with the wind pressure. This fixed condition makes it possible for every crack around the window or door to be a different width, depending on the original fitting. With so many different sized cracks, so many combinations are possible that it would be utterly impractical to test each condition separately. A method of interpolation might be used to determine the leakage for any condition but the resulting mass of figures would have no real practical value. In this work only a few conditions, such as might occur in practice, were tested. The results should give the heating and ventilating engineer some data on what to expect in ordinary practice.

In fitting casement windows and doors several factors which help determine the resistance to air leakage are taken into consideration. The hinge side must be given sufficient crack and clearance to permit free swinging and insure against binding. On the lock side or at the meeting rail, as the case may be, sufficient crack must be allowed so that the member will close easily without binding. Doors must have considerable clearance from the floor to allow them to swing freely over rugs or mats. If the rugs are thick, a large free opening results causing a draft which would probably be a large percentage of the total leakage around the door.

Table 1 gives the results of tests made on a door 3 ft. wide, 5 ft. and $9^1/_2$ in. high, having a crack perimeter of 17.6 ft. To illustrate the importance of the crack at the bottom of a door consider the leakage around a door with $^1/_{16}$ in. and with $^1/_8$ in. crack at the bottom, the other cracks being the same. From the table, the infiltration around a door with $^1/_{16}$ in. crack all around is 14.2 cu. ft. per hour per

[1] Journal of A. S.H.&V.E., Feb. and June, 1924.

Presented at the Annual Meeting of the American Society of Heating and Ventilating Engineers, Boston, January, 1925.

TABLE 1. RESULTS OF TESTS ON PLAIN DOOR

Top and lock side cracks	Clearance 1/16" Bottom 1/16" Hinge 1/16" Inf.	B.t.u.	Hinge 1/8" Inf.	B.t.u.	Bottom 1/8" Hinge 1/16" Inf.	B.t.u.	Hinge 1/8" Inf.	B.t.u.
1/16	14.2	17.9	14.9	18.8	19.3	24.3	20.0	25.2
3/32	14.6	18.4	15.5	19.5	19.7	24.8	20.6	26.0
1/8	15.2	19.15	16.3	20.5	20.3	25.6	21.4	27.0
3/16	16.8	21.2	18.3	23.1	21.9	27.6	23.4	29.5
1/4	19.0	23.9	20.9	26.4	24.1	30.4	26.0	32.1

Infiltration—cu. ft. per hour, per foot, per mile.
B.t.u. 70 deg. Temperature Difference, cu. ft., per hour, per foot, per mile.

foot of crack per mile of wind velocity. With 1/8 in. crack at the bottom and 1/16 in. cracks around the other three edges the infiltration is 19.3 cu. ft. The total leakage for the door in each case would be: 14.2 cu. ft. × 17.6 ft. = 250 cu. ft., and 19.3 cu. ft. × 17.6 ft. = 340 cu. ft. Therefore the difference, 90 cu. ft. per hour per mile of wind velocity, takes place through 3 ft. of crack at the bottom of the door, or 30 cu. ft. per foot. Since this crack is a free opening with no obstructions, the leakage through it will be proportional to the width of the crack or, in other words, an infiltration of 30 cu. ft. per hour per foot per mile of wind velocity will take place through every 1/16 in. width of crack. Therefore, in the case of the 1/8 in. crack, the infiltration would be 60 cu. ft. per hour per foot per mile of wind velocity. Subtracting the total infiltration through the bottom, or 90 cu. ft. in case of the 1/16 in. crack, from the total of 250 cu. ft. gives 160 cu. ft. through the remaining 14.6 ft. of crack around the door, or 160 divided by 146 = 10.95 cu. ft. per hour per foot, which is approximately one-third the infiltration through the same sized crack at the bottom. In many cases it is necessary that this bottom crack shall be more than 1/8 in. further increasing the infiltration.

Besides the ordinary shrinkage due to drying out and aging which takes place in all doors and windows, another factor which increases the crack must be considered in the case of doors. In warm, humid weather exposed doors sometimes swell so much, due to the greater amount of wood used in them, that they stick. When this occurs, it is usually remedied by planing off the edges of the doors. Then when the heat is turned on in the building again, the door shrinks when it dries out and consequently leaves a larger crack for air to pass through. Doors, therefore, usually have larger cracks than other openings. The same condition sometimes arises in casement windows.

The crack on the hinge side remains practically the same after the door or casement is once hung since the hinges hold it in the same position.

Table 2 gives the results obtained from tests on a double inswinging casement window, 3 ft. wide and 3 ft. 6 in. high, having a crack perimeter of 16.5 ft. The crack at the meeting rail is made greater when the window is installed so that the two members will not bind when closing the window. This is accomplished in the door by making the lock edge slightly beveled.

TABLE 2. RESULTS OF TESTS ON DOUBLE, INSWINGING CASEMENT WINDOW—PLAIN

Crack at Hinge Edge 1/16"	Clearance 1/16"				
Top and bottom cracks	1/16	3/32	1/8	3/16	1/4
Meeting rail crack	1/8	5/32	3/16	1/4	5/16
Infiltration—cu. ft. per hour per foot per mile	10.1	11.0	11.65	12.5	13.0
B.t.u. 70 deg. temp. diff.—cu. ft. per hour per foot per mile	12.7	13.85	14.7	15.7	16.35

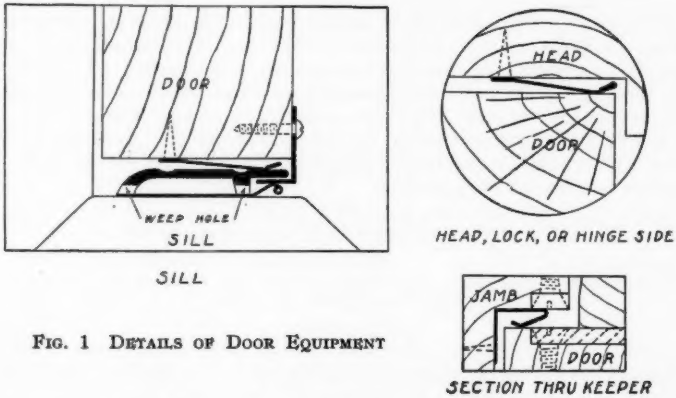


FIG. 1 DETAILS OF DOOR EQUIPMENT

A single casement window would have the same leakage as three sides of a door, including the bottom, since the construction is the same. If these values are desired, they can be figured from Table 1.

The clearance referred to in these tables is the distance between the movable member and the stops when the door or window is closed.

Weatherstripped Doors and Windows

Fig. 1 shows the details of the door equipment tested. This equipment, properly applied, shows the same leakage for all cracks and clearances. All contacts are spring contacts and their tightness depends not on the crack or clearance but on the smoothness of the surfaces. If the cracks become larger, the strip can be sprung so that it still makes contact. Table 3 shows the results of tests on this equipment.

Fig. 2 shows the details of the casement equipment tested and Table 4 shows the results obtained from the tests. The infiltration varies very little with varia-

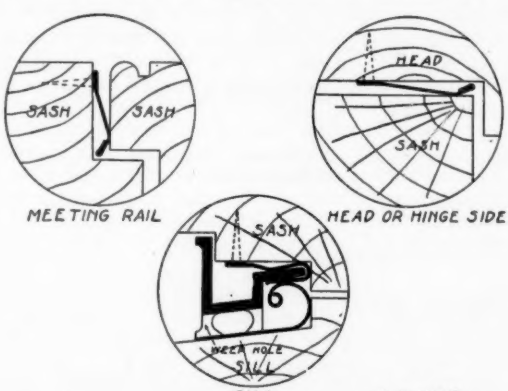


FIG. 2. DOUBLE CASEMENT EQUIPMENT

tion in the crack. The values for the smaller cracks are less than the values for the door because there is no keyhole to allow infiltration. The increase in the values as the crack increases is due to a condition present at the meeting rail. At the top of the meeting rail the weatherstrip member makes no contact on account of the crack between the two sash. This leaves a small hole through which air may pass, and as this hole becomes larger as the crack increases the infiltration becomes greater. Otherwise, the infiltration would remain the same for all cracks and clearances as is the case with the door.

TABLE 3. RESULTS OF TESTS ON WEATHERSTRIPPED DOOR

Infiltration	cu. ft., per hour, per foot, per mile	0.52
B.t.u. 70 deg. temp. diff.	cu. ft., per hour, per foot, per mile	0.655

For all cracks and clearances

Accurate data concerning infiltration and the proper application thereof are essential in the calculation of heating requirements. The most important factors in determining the amount of heating required on account of infiltration are the sizes of cracks around openings, the wind velocity, and the outside temperature. Two of these factors, namely, wind velocity and temperature, are beyond control. The third factor, crack size, can be controlled only to a very limited extent. But this slight control is acquired only at the cost of so much inconvenience as to make it impractical. To keep down the sizes of the cracks would necessitate fitting the windows or doors so tight originally that it would require considerable force to operate them. In this case the slightest swelling would make them immovable.

On the calculation of heating requirements for buildings with plain windows and doors, the part required to heat the inleaking air is a large percentage of the total requirements. An extreme condition present in any one of the three factors mentioned causes an increase in this percentage. Extreme conditions simultaneously in all three factors raises this percentage to such an extent that the heating system fails to meet this overload.

The application of weatherstripping tends to minimize the individual or combined effects of these factors. The crack size is fixed and reduced to a minimum. Controlling this one factor greatly reduces the effect of the others because it reduces the amount of heat required for the inleaking air to a small percentage of the total requirements. An extreme condition of one or more of the factors increases this percentage very slightly. Thus the variation in temperature inside the building is kept within very close limits.

TABLE 4. RESULTS OF TESTS ON DOUBLE, INSWINGING CASEMENT WINDOW—WEATHERSTRIPPED

Crack at Hinge Edge 1/16"	Clearance 1/16"				
Top and bottom cracks	1/16	3/32	1/8	3/16	1/4
Meeting rail crack	1/8	5/32	3/16	1/4	5/16
Infiltration—cu. ft. per hour per foot per mile	0.465	0.475	0.500	0.580	0.690
B.t.u. 70 deg. temp. diff. per foot per mile—cu. ft. per hour	0.585	0.598	0.630	0.730	0.870

No. 717

EFFECTIVE TEMPERATURE WITH CLOTHING[1]

By C. P. Yaglou[2] (Member), and W. Edw. Miller[3] (Non-Member)

Pittsburgh, Pa.

Equivalent Conditions of Temperature, Humidity and Air Movement, Determined with Individuals Normally Clothed

Introduction

IN THE fundamental development of the effective temperature scale the influence of routine factors such as clothing and muscular activity was eliminated in order that a separate study could be made thereof. The effect of these factors is now being studied in the order of their importance, and the object of this paper is to present a normal scale of effective temperature for ordinary conditions of life, as determined with individuals normally clothed and slightly active.

Considerable effort has been made to present the final results of this investigation in a simple and practical chart, especially for those who are not proficient in the use of psychrometric charts and complicated figures.

Clothing aids the human body in maintaining its constant temperature. Air is entangled and rendered stationary within its cellular structure and between its layers, thus insulating the body against heat loss. The loss of heat by radiation and convection is reduced considerably through a decrease in the surface temperature, but the heat loss by evaporation may, under certain conditions, increase because of the greater surface afforded by the clothing.

The rate at which clothing transfers heat through it depends upon the material, condition of same with respect to moisture, thickness and size of its meshes. When dry, cotton or woolen clothes of the same thickness and size of mesh are equally good, but when wet, woolen clothes prevent heat loss much better than cotton. In general, it can be stated that other things being equal, the rate at which clothing transfers heat depends upon the amount of air within its meshes and between its layers.

It is obvious that it will be impractical in this particular problem to study

[1] Work carried on in conjunction with the U. S. Bureau of Mines at its Pittsburgh Experiment Station.

[2] Research Engineer, A.S.H.&V.E. Research Laboratory.

[3] Research Assistant, A.S.H.&V.E. Research Laboratory.

Presented at the Annual Meeting of the American Society of Heating and Ventilating Engineers, Boston, January, 1925.

the effect of different kinds of clothing, but if ordinary garments of medium thickness and size of mesh are chosen in the experiments, any reasonable departure from this average in actual practice will not affect the results to any great extent.

General Method

In the previous experiments on effective temperature described in previous issues of the JOURNAL,[4] the subjects were stripped to the waist, similar to many industrial workers who are exposed to high temperatures. The subjects of the experiments now to be described were clothed in garments suitable for all seasons of the year. The clothing worn consisted of light weight union cotton underwear, madras shirt with collar attached, a three-piece medium-weight woolen suit of medium-size mesh, cotton socks, and shoes.

The activities of the subjects during all tests were those incidental to carrying out the experiments, involving walking from one chamber into the other, manipulating valves for controlling the temperature conditions, taking readings and mentally concentrating on the comparative feeling of warmth of the two conditions.

In principle, the same procedure was followed as in the previous experiments to determine what conditions were equivalent. Briefly, the method consisted in finding a condition of comparatively high dry-bulb temperature and low humidity in one chamber, which gave the same feeling of warmth as a saturated condition with lower temperature in the other chamber. In the experiments with air movement, the humidity in the two chambers was kept the same, but the dry-bulb temperature in the velocity chamber was, as a general rule, higher than that in the still-air chamber thus counterbalancing the cooling effect of wind. (For further details and tunnel arrangement the reader is referred to the articles mentioned previously.)

Preliminary to every experiment, the subjects upon entering the test chamber spent about half an hour in preparatory work, to allow time for the clothing to attain a temperature equilibrium with the room air. In the very high temperatures the clothing was exposed to the hot air for some time before putting it on. Changes in the room conditions were gradual, and the field covered in a single day with full range of humidity did not exceed 20 deg. in dry-bulb temperature.

Considerable difficulty was experienced by the subjects when first starting this series of experiments in comparing the temperature conditions with a reasonable degree of accuracy. There was an inclination to judge conditions by the warmth sensed by the face and hands only, because the covered parts of the body did not seem to feel small temperature variations. The preliminary data thus obtained were practically the same as those already obtained with the subjects stripped to the waist. Eventually, however, the subjects were adapted to the new routine, and became proficient in analyzing their feelings in terms of the "overall" warmth sensed by the entire body. The shoulders and chest areas of the body were found to be most sensitive of the covered parts.

Results of Experiments

Some experimental data are presented in Figs. 1 and 3 to give an idea of the magnitude of the error involved in determining the equivalent conditions, and also to

---

[4] Lines of Equal Comfort, by F. C. Houghten, and C. P. Yagloglou, JOURNAL of the A.S.H.&V.E., March, 1923. Cooling Effect on Human Beings Produced by Various Air Velocities, *Ibid.*, February, 1924. Effective Temperature Applied to Industrial Ventilation Problems, by C. P. Yaglou, and W. E. Miller, *Ibid.*, July 1924.

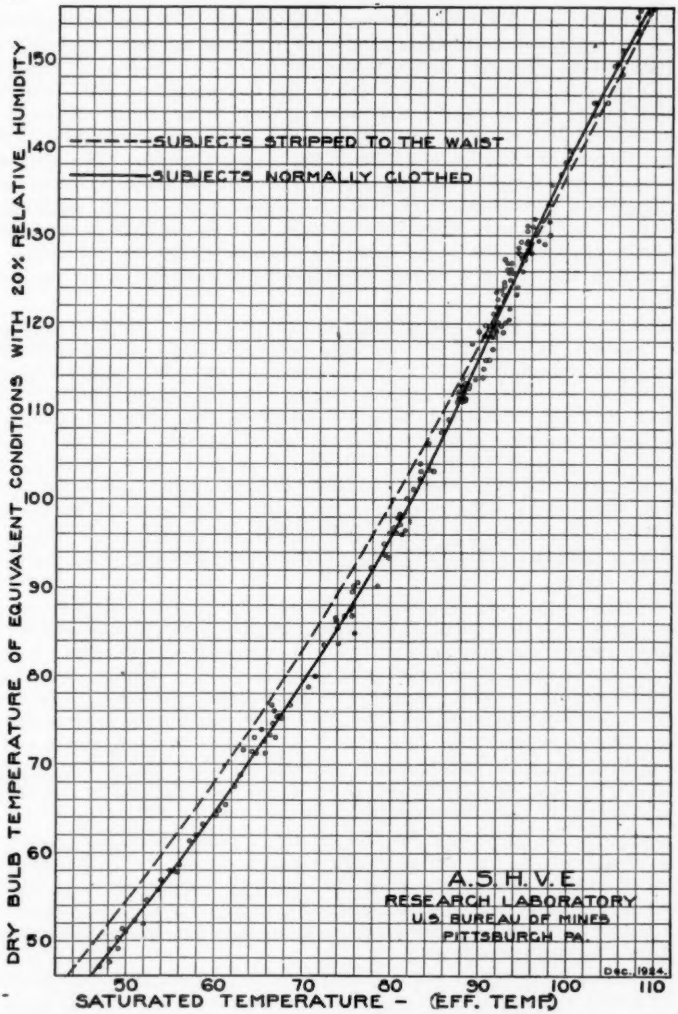


FIG. 1. RESULTS OF STILL AIR EXPERIMENTS. SOLID CURVE SHOWS SATURATED ATMOSPHERIC CONDITIONS FOUND TO BE EQUIVALENT TO OTHERS WITH 20 PER CENT RELATIVE HUMIDITY BY SUBJECTS NORMALLY CLOTHED. DOTTED CURVE GIVES SIMILAR CONDITIONS FOR INDIVIDUALS STRIPPED TO THE WAIST

indicate the effect of clothing in comparison with the previous results obtained with subjects stripped to the waist. The temperature range covered in the present work was from 33 to 165 deg. dry bulb and from 26 to 115 deg. wet bulb.

Fig. 1 shows part of the data obtained with still air. The horizontal scale represents the saturated temperature conditions in one of the test chambers, while the vertical scale gives the dry-bulb temperature of their equivalent conditions with 20 per cent relative humidity that is, the temperature to which the other chamber had to be brought for the two conditions to feel equally warm to the human body. Not all of these data, however, were actually determined with exactly 20 per cent relative humidity in the second chamber. The equivalent conditions as determined in the experiments were first plotted on a psychrometric chart, and lines were drawn through the homogeneous points. By duplicate experiments these lines were found to be straight, and the dry-bulb temperature of their intersection with the 20 per cent relative humidity line was taken from the chart and plotted in Fig. 1 against the saturated temperature of the equivalent conditions.

The solid curve in Fig. 1 averages the experimental points, and gives the equivalent conditions with the subjects normally clothed, while the dotted curve shows the corresponding conditions for subjects stripped to the waist. Both curves show the same general tendencies and characteristics. For temperatures below 100 deg. dry bulb they are parallel indicating a constant relative reduction in heat loss when clothing is worn. It will be observed that a saturated condition of 65 deg. is equivalent to one of 71.7 deg. dry bulb with 20 per cent relative humidity when clothing is worn, and to another of the same humidity as the latter but of 75 deg. dry-bulb temperature, when stripped to the waist. In other words, for the same effective temperature value of 65 deg. and with 20 per cent relative humidity, the dry-bulb temperature in the former case (with normal clothing) is 3.3 deg. lower than in the latter.

In the region near body temperature, where the heat loss by radiation and convection approaches zero, sensible perspiration begins. Clothing increases the loss of heat by evaporation due to the greater surface exposed, as a result of which the lower curve converges toward the upper, until at 94 deg. effective temperature they cross each other. At this condition it is immaterial whether clothing is worn or not, but when this temperature is exceeded the reversal in the order of the two curves indicates the beneficial effect of clothing in increasing evaporation and decreasing heat gained by radiation and convection. Therefore, in industrial operations that require exposure to high temperatures, accompanied by low humidities, workers will be better off if their entire body is covered with clothing.

A psychrometric chart with the effective temperature lines superimposed, for still-air conditions and with normal clothing is shown in Fig. 2. In this figure the relation of effective temperature to dry and wet-bulb temperature, and moisture content of the air can be studied in detail. Although a much greater temperature range was employed in the experiments, for clearness in presentation the extent of the chart is reduced considerably.

A comparison of this chart with the one presented in the March, 1923 JOURNAL, for individuals stripped to the waist, will show clearly the effect of clothing. It will be observed that clothing increases considerably the slope of the effective-temperature lines below 94 deg. effective temperature and decreases their slope above this temperature. In other words, for ordinary temperatures dry bulb becomes a much more important factor in the comfort of a clothed person than wet-bulb temperature, a fact that justifies to a certain extent the original belief in dry-bulb temperature as being the sensible temperature.

Although at 65 deg. effective temperature it was found previously that for subjects stripped to the waist 1 deg. of dry-bulb temperature is equivalent to 0.9

deg. of wet-bulb, it is significant to note that for subjects normally clothed, 1 deg. of dry bulb is equivalent to 2 deg. wet bulb. Therefore, under ordinary conditions of temperature and clothing, an increase of 1 deg. in the dry-bulb temperature of the air should be accompanied by a decrease of 2 deg. in wet bulb, in order that the warmth of the condition may remain unchanged.

This influence of clothing may be explained by the transfer of heat through cellular hygroscopic materials. Moist air is a better vehicle of heat transfer than dry air, therefore at high humidities the heat conductivity of clothing is increased appreciably. On the other hand, dry heat penetrates very slowly into and through the alternate layers of clothing, so that the latter affords better protection to the human body against heat loss at low than at high humidities. This is expressed by an increase in the slope of the effective temperature lines as shown in Fig. 2.

It is of interest to note in this connection that the clothed body simulates a dry-bulb thermometer at a temperature of 46 deg., while this was found to take place with subjects stripped to the waist at 32 deg. Below this temperature the higher the humidity the cooler the condition. This fact brings us nearer to common experience in observing the difference between dry and damp weather at temperatures considerably above freezing. As a matter of fact, there is always outdoors a certain amount of air movement, and it will be seen later that the greater the air velocity the higher the temperature at which the clothed body assumes the thermal properties of a dry-bulb thermometer.

In Fig. 3 is presented all the data collected in a series of saturated tests with 300 ft. per minute air velocity. The vertical scale gives the saturated temperature conditions with 300 ft. velocity that were found to be equivalent to the still air conditions of the horizontal scale. For example, a saturated condition of 65 deg. with still air was found to be equally warm as one of 71 deg. with a velocity of 300 ft. per minute. The difference of 6 deg. represents the cooling effected by the movement of the air. The dotted curves show that, with the subjects stripped to the waist, the corresponding difference is about 10.5 deg. Therefore clothing at this particular condition reduces the cooling by 43 per cent.

It will be observed that these curves show the same characteristics as those in Fig. 1. At 90 deg. effective temperature, the two curves cross each other, and the beneficial effect of clothing at temperatures above that of the body is shown by the reversal in the order of the two curves. Clothing therefore protects the human body against heat loss at low temperatures and against heat gain when the outside temperature exceeds that of the body. This protection, however, becomes smaller and smaller as the velocity of the air increases.

Similar data have been obtained for humidities of approximately 80 per cent, 50 per cent and 15 per cent, and for air velocities of 150 ft., 300 ft., and 500 ft. per minute, from which the slope and curvature of the effective temperature lines were determined. A psychrometric chart with the lines superimposed for a velocity of 300 ft. per minute is shown in Fig. 4. The lines are straight over the entire range of temperature, although with the subjects stripped to the waist there was a temperature zone in which the lines were distinctly curved. Probably when clothing is worn the body is not as sensitive to small humidity changes as it is when stripped to the waist.

A comparison of Figs. 4 and 2 will disclose the increased slope of the effective-temperature lines with velocity. Dry-bulb temperature becomes a much more predominent indicator of bodily comfort, and the body approaches the thermal properties of a dry-bulb thermometer at a temperature of 56 deg. for an air velocity of 300 ft. per minute.

The dotted line at the top of the figure passes through all the neutral conditions at which the effect of the wind becomes zero. It constitutes a border line that separates the cooling zone below it from the heating zone above it.

Similar psychrometric charts have been prepared, one for each of the velocities investigated, but the one shown here is representative of the group. In general, it can be stated that as the velocity increases, dry-bulb temperature becomes more important than wet-bulb temperature.

Efforts to devise a single practical chart that will include all the facts pertaining to the laws governing the thermal properties of the human body, and by means of

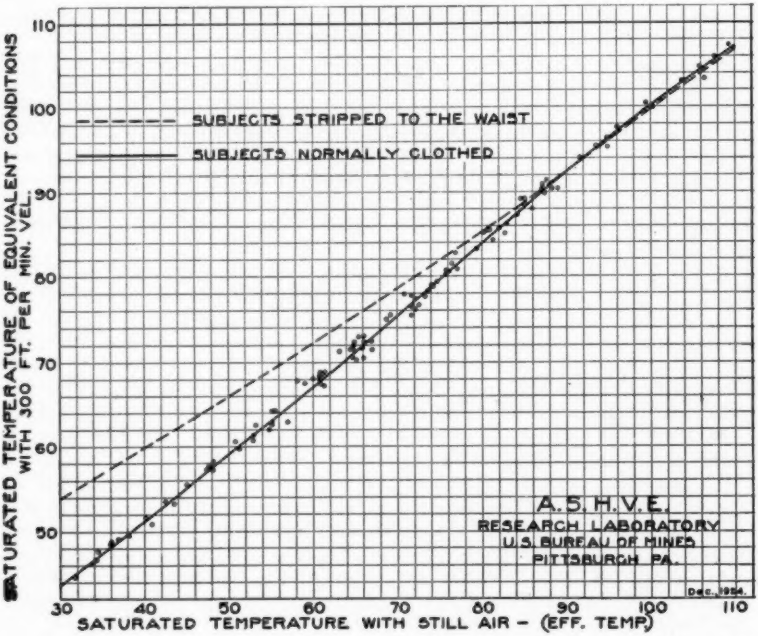


FIG. 3. RESULTS OF EXPERIMENTS WITH 100% RELATIVE HUMIDITY AND WITH 300 FT. PER MINUTE VELOCITY. CURVES GIVE EQUIVALENT SATURATED CONDITIONS FOR SUBJECTS BOTH NORMALLY CLOTHED AND STRIPPED TO THE WAIST

which the effective temperature of any condition within the experimental range could be easily determined, resulted in the development of the "human thermometric chart" shown in Fig. 5. The chart is principally composed of two vertical and uniform scales; the dry-bulb temperature scale on the left and the wet-bulb scale on the right; these are the two values actually measured in practice. The long curved lines extending diagonally across the chart are lines of equal air velocity spaced in intervals of 100 ft. per minute, as shown by the scale on the left edge of the diagram. The effective-temperature lines are represented by the short cross lines, and join together all equivalent conditions of dry-bulb temperature, wet bulb, and velocity of air.

As an example in the use of the chart, assume a condition of 76 deg. dry bulb and 62 deg. wet bulb, with a velocity of 100 ft. per minute. Lay a straight edge on the proper temperatures upon the two vertical scales, as shown in the figure by the dotted line *A-B*. Its intersection with the 100 ft. velocity line gives 69 deg. for the effective temperature of the condition. The point at which the dotted line crosses the zero velocity curve gives 70.4 deg. for the effective temperature of the condition with still air. The difference of 1.4 deg. between the two values is the cooling produced by the movement of the air. If the velocity of the air was 500 ft. per minute instead of 100 ft., the effective temperature would be 64.6 deg., and the corresponding cooling 5.8 deg. effective temperature.

Having thus determined the effective temperature of the condition, any combination of dry and wet-bulb temperatures that will intersect any of the velocity curves on the 69 degree effective-temperature line will be equivalent to the given condition. For instance, a dry-bulb temperature of 82.0 deg. and wet-bulb temperature of 58.0 deg. with an air velocity of 300 ft. per minute, will be equivalent to the original condition, and also equivalent to a saturated one of 71 deg. with a velocity of 100 ft. per minute.

If it is required to find the velocity necessary to reduce the given atmospheric condition to 66 deg. effective temperature, follow line *A-B* to the left until it crosses the comfort line. By interpolation it will be found that the velocity of air should be increased to 340 ft. per minute for maximum comfort.

For humidities between 20 and 100 per cent this chart is as accurate as any psychrometric chart with the effective-temperature lines superimposed, and values obtained from the first agree within two-tenths of a degree with those in Figs. 2 and 4. For humidities between 20 and 0 per cent the maximum variation is about $^1/_2$ degree.

Fig. 5 affords a complete formulation of the laws of cooling of the human body. The proximity of any point to the dry- and wet-bulb temperature axes indicates the superiority of one temperature over the other in determining human comfort. For ordinary temperatures the points are nearer to the dry-bulb temperature axis, while for high temperatures, when sensible evaporation comes into play, the upper part of the diagram approaches the wet-bulb axis. A vertical line drawn halfway between the two temperature scales will intersect the various velocity curves at points where dry- and wet-bulb temperatures are of equal importance. For instance, with still air this occurs at an effective temperature of about 77 deg., while with a velocity of 500 ft. per minute it occurs at a saturated temperature of 86.0 deg., or at an effective temperature of 80.7 deg.

It will be observed that in the lower part of the diagram the various velocity lines intersect the dry-bulb axis at temperatures at which the effect of wet bulb, or humidity of the air, is eliminated completely. Below these temperatures the divergence of the velocity lines away from the dry-bulb axis and the change in their curvature indicate a reversal in the effect of humidity; the higher the humidity the cooler the condition, and vice versa.

At high temperatures the velocity curves converge toward each other, until at body temperature they all meet at a common point called the "neutral point" in atmospheric conditions. Above this point the reversal in the order of the velocity curves with reference to the still-air curve shows the heating effect of wind upon the human body, and its increase with velocity. For temperatures higher than 120 deg. dry bulb, but not exceeding 170 deg., the scale may be extended beyond the limit of the chart and the values determined in the usual manner as long as the effective temperature is below 110 deg.

TABLE 1. FOR DETERMINING EFFECTIVE TEMPERATURE

Subjects Normally Clothed and Slightly Active
Effective Temperature
Velocity of Air in Feet per Min.
Relative Humidity

Dry Bulb Temp. °F.	Still Air			50 ft.			100 ft.			200 ft.			300 ft.			500 ft.			700 ft.		
	100%	50%	20%	100%	50%	20%	100%	50%	20%	100%	50%	20%	100%	50%	20%	100%	50%	20%	100%	50%	20%
30	30	31.8	32.7																		
35	35	36.3	36.9		33.3	34.0		30.4	31.3												
40	40	40.8	41.1	37.2	38.2	38.6	34.4	35.8	36.3		31.1	32.0									
45	45	45.1	45.2	42.6	43.1	43.2	40.0	40.8	41.2	35.6	36.7	37.5	31.9	33.7	34.5			30.8			
50	50	49.4	49.2	48.0	47.8	47.6	45.7	45.9	45.9	41.7	42.4	42.7	38.4	39.5	39.9	34.0	35.9	36.6	30.8	33.6	34.8
52	52	51.1	50.8	50.1	49.7	49.3	48.0	47.8	47.7	44.2	44.6	44.7	41.0	41.7	42.0	36.8	38.3	38.8	33.8	36.0	37.1
54	54	52.9	52.4	52.2	51.5	51.0	50.2	49.7	49.5	46.7	46.7	46.7	43.6	44.0	44.2	39.6	40.7	41.0	36.7	38.5	39.5
56	56	54.6	53.9	54.3	53.2	52.6	52.4	51.7	51.2	49.1	48.8	48.6	46.2	46.2	46.2	42.4	43.0	43.2	39.7	41.0	41.8
58	58	56.2	55.3	56.4	54.9	54.1	54.7	53.5	52.9	51.5	50.8	50.5	48.7	48.4	48.2	45.1	45.3	45.4	42.6	43.5	44.0
60	60	57.9	56.8	58.5	56.7	55.7	56.9	55.4	54.6	54.0	52.8	52.3	51.3	50.6	50.2	47.8	47.7	47.6	45.4	45.9	46.2
61	61	58.7	57.6	59.6	57.6	56.5	58.0	56.3	55.4	55.2	53.8	53.1	52.5	51.6	51.1	49.1	48.8	48.6	46.8	47.1	47.3
62	62	59.6	58.3	60.6	58.5	57.3	59.1	57.2	56.2	56.4	54.8	54.0	53.7	52.7	52.1	50.5	50.0	49.6	48.2	48.3	48.3
63	63	60.3	59.0	61.7	59.3	58.0	60.2	58.1	57.0	57.6	55.8	54.9	55.0	53.7	53.0	51.8	51.1	50.6	49.6	49.5	49.4
64	64	61.1	59.7	62.7	60.2	58.8	61.3	59.0	57.8	58.8	56.8	55.7	56.2	54.7	54.0	53.1	52.2	51.7	51.0	50.7	50.4
65	65	62.0	60.5	63.8	61.0	59.5	62.4	59.9	58.6	60.0	57.7	56.6	57.5	55.7	55.0	54.5	53.3	52.7	52.4	51.8	51.4
66	66	62.8	61.2	64.8	61.8	60.3	63.5	60.7	59.4	61.1	58.7	57.5	58.7	56.8	55.9	55.8	54.4	53.7	53.8	53.0	52.4
67	67	63.6	61.9	65.8	62.7	61.0	64.6	61.6	60.2	62.3	59.6	58.3	60.0	57.8	56.8	57.1	55.5	54.6	55.1	54.1	53.4
68	68	64.4	62.6	66.9	63.5	61.7	65.7	62.5	60.9	63.4	60.6	59.2	61.2	58.8	57.7	58.4	56.5	55.6	56.4	55.2	54.5
69	69	65.2	63.3	67.9	64.4	62.5	66.8	63.4	61.7	64.6	61.6	60.0	62.5	59.8	58.6	59.7	57.6	56.6	57.7	56.3	55.5
70	70	66.0	64.0	68.9	65.2	63.2	67.9	64.3	62.5	65.7	62.5	60.8	63.7	60.8	59.5	61.0	58.6	57.5	59.1	57.4	56.5
71	71	66.8	64.7	70.0	66.0	63.9	69.0	65.1	63.2	66.9	63.4	61.7	65.0	61.8	60.3	62.2	59.7	58.5	60.4	58.5	57.4
72	72	67.6	65.4	71.0	66.8	64.6	70.0	66.0	63.9	68.0	64.3	62.5	66.2	62.7	61.2	63.5	60.7	59.4	61.7	59.5	58.3
73	73	68.4	66.0	72.1	67.6	65.3	71.1	66.8	64.6	69.2	65.2	63.3	67.4	63.7	62.0	64.7	61.8	60.3	63.0	60.5	59.3
74	74	69.2	66.7	73.1	68.4	66.0	72.2	67.7	65.4	70.3	66.1	64.1	68.6	64.7	62.9	66.0	62.8	61.2	64.3	61.5	60.2
75	75	70.0	67.4	74.2	69.2	66.7	73.2	68.5	66.1	71.4	67.0	64.9	69.8	65.7	63.7	67.3	63.8	62.2	65.6	62.6	61.1
76	76	70.7	68.0	75.2	70.0	67.4	74.3	69.3	66.8	72.5	67.9	65.6	71.0	66.6	64.6	68.5	64.8	63.1	66.8	63.6	62.0
77	77	71.5	68.7	76.2	70.8	68.1	75.4	70.1	67.5	73.7	68.8	66.4	72.2	67.5	65.4	69.7	65.8	63.9	68.0	64.6	62.9
78	78	72.3	69.4	77.2	71.6	68.8	76.4	70.9	68.2	74.8	69.7	67.2	73.3	68.4	66.2	71.0	66.8	64.8	69.3	65.7	63.8
79	79	73.1	70.0	78.2	72.4	69.5	77.5	71.7	68.9	76.0	70.5	67.9	74.5	69.4	67.0	72.2	67.7	65.6	70.5	66.7	64.7
80	80	73.9	70.7	79.2	73.1	70.1	78.5	72.6	69.6	77.1	71.3	68.6	75.6	70.3	67.8	73.4	68.7	66.5	71.7	67.6	65.6
82	82	75.4	71.9	81.3	74.7	71.4	80.6	74.1	70.9	79.3	73.0	70.0	78.0	72.0	69.3	75.8	70.6	68.1	74.3	69.6	67.3
84	84	76.9	73.2	83.3	76.3	72.7	82.7	75.7	72.2	81.5	74.7	71.4	80.2	73.8	70.7	78.2	72.4	69.7	76.7	71.4	68.9
86	86	78.4	74.4	85.4	77.8	74.0	84.8	77.3	73.6	83.7	76.3	72.8	82.6	75.5	72.1	80.6	74.2	71.2	79.3	73.2	70.5
88	88	79.9	75.6	87.5	79.3	75.2	86.9	78.7	74.8	85.9	77.9	74.1	84.8	77.1	73.5	83.1	76.0	72.7	81.9	75.0	72.1
90	90	81.4	76.8	89.5	80.8	76.4	89.0	80.3	76.0	88.1	79.5	75.4	87.1	78.8	74.8	85.6	77.6	74.1	84.5	76.8	73.5
92	92	82.8	77.9	91.6	82.4	77.6	91.1	81.8	77.2	90.4	81.0	76.6	89.5	80.4	76.1	88.1	79.2	75.5	87.3	78.6	74.9
94	94	84.2	79.1	93.7	83.8	78.7	93.3	83.3	78.4	92.7	82.5	77.8	92.0	81.9	77.4	90.8	80.8	76.9	90.1	80.2	76.3
96	96	85.6	80.2	95.8	85.2	79.8	95.5	84.7	79.5	95.1	84.0	79.0	94.5	83.4	78.6	93.7	82.4	78.0	93.2	81.8	77.5
98	98	87.0	81.3	97.9	86.6	81.0	97.7	86.1	80.6	97.4	85.5	80.2	97.2	84.9	79.8	96.7	84.0	79.2	96.4	83.4	78.8
100	100	88.4	82.4	100.0	88.0	82.0	100.0	87.6	81.7	100.0	86.9	81.2	100.0	86.3	80.9	100.0	85.5	80.4	100.0	85.0	80.0
105	105	91.6	85.0	105.2	91.4	84.6	105.5	91.1	84.3	106.1	90.5	83.9	106.9	90.0	83.6	108.4	89.4	83.2		89.0	82.8
110	110	95.1	87.4		94.9	87.0		94.6	86.8		94.2	86.5		93.9	86.1		93.3	85.7		93.1	85.4
115	115	98.6	89.6		98.5	89.4		98.3	89.2		98.1	88.9		98.0	88.5		97.6	88.2		97.4	87.9
120	120	102.2	91.8		102.3	91.6		102.3	91.5		102.3	91.1		102.5	90.9		102.7	90.6		102.8	90.3
125	125	106.0	93.9		106.3	93.8		106.6	93.6		107.1	93.4		107.7	93.1		108.9	92.9			92.5

Considering next the efficiency of air movement in cooling the human body, it will be noted in Fig. 5, that low velocities are more efficient than high. The distance between any two consecutive velocity lines is a function of the amount of cooling produced by an increase of 100 ft. per minute in the velocity of the air. Therefore above 300 ft. per minute the efficiency of air movement falls off considerably, and it will be inefficient in practice to use velocities greater than 300 ft. per minute.

In the application of effective temperature to practical problems, where air of high temperature and comparatively low humidity is cooled, and at the same time saturated with moisture by passing it through a humidifier, Fig. 5 affords a simple means of studying in detail the cooling that will result from saturation and movement of the air. In problems involving cooling or heating of the air at constant dew point, when a detailed study is contemplated, it will be desirable to introduce in the chart a scale of moisture content of the air. For clearness in reproduction this scale is not shown here, because with the chosen distance between the two vertical axes it falls outside the limits of the chart.

Table for Determining Effective Temperature

As a matter of recording the results of the experiments in the Society's publications, the data are also presented in Table 1, by means of which the effective temperature of a condition can be determined from the dry-bulb temperature, relative humidity, and velocity of the air. This table is very similar to the one for subjects stripped to the waist appearing in the July, 1924, JOURNAL, and therefore a detailed explanation of it will be unnecessary here. In so much as in the present experiments the effective temperature lines were found to be straight, their values are given in the table for the ordinary humidity of 50 per cent and for two extreme humidities, of 100 per cent and 20 per cent. Interpolation for humidities between gives results accurate within one or two tenths of a degree effective temperature. For the same dry-bulb temperature, relative humidity, and velocity of air, the difference between the values in this table and those given in the JOURNAL mentioned represents the effect of clothing in reducing the cooling power of the atmosphere.

General Discussion and Practical Conclusions

The Research Laboratory has thus presented to the members of the Society and the industrial world in general two fundamental investigations on effective temperature of vital importance to human comfort and efficiency: In the first study, the basic scale of effective temperature has been developed in a simple form and under physiological conditions of maximum sensitivity of the skin toward external temperature conditions. This basic scale is applicable to all similar cases of bodily exposure, such as are found in many industries where the temperature conditions are abnormally high and necessitate the removal of superfluous clothing.

In the second investigation ordinary clothing was worn by the subjects of the experiments, so that a "normal" scale of effective temperature was derived. This second scale is applicable to ordinary conditions and occupations of life, and emphasizes the importance of the first study in showing the extent to which clothing reduces the cooling effect of wind.

The Research Laboratory is as yet unprepared to discuss in detail the industrial use of equivalent conditions and the extent to which air movement will improve working conditions in cases where strenuous muscular work is performed. But any principle, or method, the use of which results in a decided improvement in the

working conditions of moderate activity, will unquestionably produce some improvement when hard muscular work is done. As a matter of fact, work increases heat loss from the human body, so that a greater cooling effect may be expected of a given velocity when hard work is done.

Effect of Clothing

To indicate the large extent to which clothing decreases the cooling power of wind in such temperatures as are frequently met with in actual practice, Fig. 6 has been prepared with velocity of air as abscissas and cooling in degrees effective

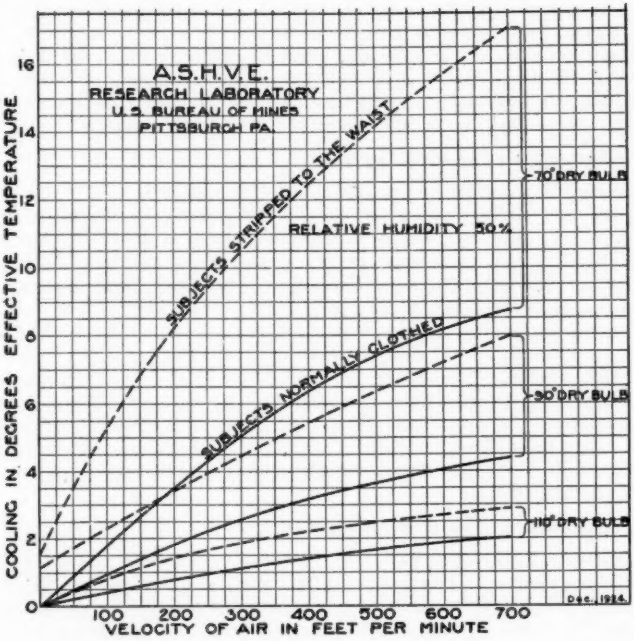


FIG. 6. CHART SHOWING THE EXTENT TO WHICH CLOTHING REDUCES THE COOLING EFFECT OF WIND AT REPRESENTATIVE TEMPERATURES

temperature as ordinates. The three sets of curves give the relative amount of cooling for three different dry-bulb temperatures, but for the same relative humidity of 50 per cent, and for subjects both normally clothed and stripped to the waist, as indicated in the figure. At the ordinary dry-bulb temperature of 70 deg., a velocity of 300 ft. per minute produces a cooling of about 5 deg. on the human body when clothed, and 10.5 deg. when stripped to the waist. Therefore clothing under these conditions reduces the cooling by 52 per cent. Approximately the same percentage reduction holds true for velocities higher than 300 ft., but for lower velocities the decrease in cooling effect is still greater. With a velocity of 100 ft. per minute, the corresponding decrease in cooling at 70 deg. is 66 per cent. As the

temperature of the air increases the percentage reduction in cooling decreases slightly, varying from a maximum of 50 per cent to a minimum of 40 per cent.

In general, it can be stated that normal clothing at ordinary humidity halves the cooling effect of wind as compared with that obtained when only light work trousers, socks, and shoes are worn. The importance of stripping to the waist is therefore apparent when high cooling power of the air is desired, provided that the temperature conditions do not exceed certain perscribed limits. On the other hand, clothing in a number of instances is advantageous in increasing loss of heat by evaporation and decreasing heat gain by radiation and convection. This is particularly true when very high dry-bulb temperatures are accompanied by humidities below 20 per cent, and when the wet-bulb temperature is well below 99 deg. In fact, man is capable in such cases of increasing evaporation materially by suitable clothing. An elastic cotton union suit which will cover tightly as much of the body's surface as possible constitutes an ideal outfit for exposure to hot atmospheres. The excessive perspiration, most of which runs down and off the body in the case of a man stripped to the waist, and thus rendered unavailable for evaporation, is absorbed by the cotton garment and distributed uniformly over the entire surface by capillary action. This method of exposure eliminates the burning effect upon the corner areas of the body which become dry, while other parts of the body's surface are cooled adequately by the evaporation of abundant perspiration.

Clothing, also, should never be removed when the wet-bulb temperature exceeds the temperature of the body. The air in such cases should be kept as still as possible, and the more the clothing the greater the insulation against transfer of heat from the air to the body.

In keeping with the best information available, the experimental data collected in a previous study of the "comfort zone"[5] is converted to the normal scale of effective temperature and the zone superimposed on Fig. 5, as shown by the dotted area. It will be observed that when referred to the "normal" scale the comfort zone is included between the 63 and 71 deg. effective-temperature lines, which constitute, respectively, the lower and upper boundaries of the zone. Strictly speaking, the "comfort line" is represented by the 66 deg. effective-temperature line, which for the range of humidity employed in the comfort experiments corresponds to a basic value of 64.5 effective temperature. Thus it is left to the judgment of the heating and ventilating engineer to shift the comfort line back or forth according to seasonal variations in the clothing worn. If lighter clothing is worn in summer than in winter, the comfort line will approach the lower boundary of the zone in winter and the upper boundary in summer.

[5] Determination of the Comfort Zone, by F. C. Houghten and C. P. Yaglou, JOURNAL A.S.H.&V.E., September, 1923.

No. 718

WORK TESTS CONDUCTED IN ATMOSPHERES OF HIGH TEMPERATURES AND VARIOUS HUMIDITIES IN STILL AND MOVING AIR

By W. J. McConnell[1] (Non-Member) and C. P. Yaglou[2] (Member)

PITTSBURGH, PA.

THE study of the physiological effects of various temperatures and humidities on human beings at rest in still and in moving air has already been carried out jointly by the Research Laboratory of the Society, the U. S. Bureau of Mines and the U. S. Public Health Service, and reported in previous papers.[3]

The object of this paper is to present the results of experiments conducted with subjects doing measurable amounts of work at a constant rate in still and in moving air of various temperatures and humidities.

Procedure

The procedure followed in the present series of experiments was of necessity somewhat different from that used in the experiments where the subject rested. The preliminary rest period of two hours before exposure to the test conditions was carried out as in the previous experiments. Experience has shown that preliminary rest is absolutely necessary and a glance at Table 1 will demonstrate its value. The table gives the rectal temperature, mouth temperature and pulse rates of subjects at the beginning and at the end of the rest period in the last six experiments of this series.

It will be noted that the preliminary activities of the subjects frequently caused a rise in temperature and pulse rate, which dropped to a more normal reading after a short period of rest.

The rectal temperature and pulse rate invariably dropped. The variations in the rectal temperature before rest were between 100.2 and 98.2 deg. fahr. and after the rest period between 99.7 and 97.8 deg., while the respective pulse rates ranged be-

---

[1] Past Asst. Surgeon (R), U. S. Public Health Service, and Surgeon, U. S. Bureau of Mines.
[2] Research Engineer, A.S.H.&V.E., Research Laboratory.
[3] The Effects of High Temperatures and Humidities, by W. J. McConnell, *The Nations' Health*, vol. 4, Oct. 1922, pp. 616–617. Some Physiological Reactions to High Temperatures and Humidities, by W. J. McConnell and F. C. Houghten, Journal A.S.H.&V.E., March, 1923, pp. 131–164. Further Study of Physiological Reactions, by W. J. McConnell, F. C. Houghten and F. M. Phillips, Journal A.S.H.&V.E., Sept. 1923, pp. 507–514. Air Motion-High Temperatures and Various Humidites—Reactions on Human Beings, by W. J. McConnell, F. C. Houghten and C. P. Yaglou, Journal A.S.H.&V.E., Mar. 1924, pp. 199–224.

Presented at the Annual Meeting of the American Society of Heating and Ventilating Engineers, Boston, January, 1925.

tween 104 and 72 and 84 and 60. The mouth temperature frequently rose after the rest period. This condition arises from the subject entering a comfortable room after walking out-doors in cold weather. Where this condition did not exist, the mouth temperature likewise dropped. Before the rest period the mouth temperatures ranged between 98.7 and 97.0 deg.; after the rest period, between 98.2 and 96.8 deg.

On entering the test chamber, half of the subjects worked in one chamber at still air and the other half worked in the other chamber maintained at the same temperature and humidity as the first, but with an air velocity of 350 ft. per min. The air struck the back of their bodies at right angles. Strictly speaking, the velocity in

TABLE 1. PHYSIOLOGICAL MEASURES OF SUBJECTS AT BEGINNING AND END OF REST PERIOD

Test No.	Subjects	Rectal Temperature Beginning	Rectal Temperature End	Mouth Temperature Beginning	Mouth Temperature End	Pulse Rate Beginning	Pulse Rate End
35	McChesney	99.8	99.4	98.0	98.2	96	84
	Milliron	99.0	98.0	97.8	96.8	78	72
	Bishop	98.5	98.0	97.4	97.3	78	70
	Ferguson	99.0	99.0	98.0	98.0	84	78
	McChesney	99.9	99.7	98.7	97.8	104	76
36	Milliron	98.7	98.0	96.6	97.8	87	72
	Bishop	99.4	99.2	97.3	98.0	74	74
	Ferguson	99.0	99.0	97.4	98.0	72	70
	McChesney	99.9	99.2	98.1	97.4	92	76
37	Milliron	99.0	98.4	97.9	97.7	90	80
	Bishop	99.0	99.0	98.1	98.0	72	70
	Ferguson	99.0	98.6	97.4	98.0	86	74
	McChesney	99.2	99.3	98.1	98.0	84	80
38	Milliron	98.6	98.2	97.6	97.8	84	84
	Ferguson	99.2	98.6	97.4	97.6	86	74
	McChesney	99.4	98.6	97.8	97.3	84	60
39	Milliron	99.8	98.2	97.6	98.0	84	72
	Bishop	99.8	99.0	98.6	98.0	96	80
	Ferguson	100.2	98.6	98.2	98.2	80	70
	McChesney	99.8	99.0	98.0	97.3	100	80
40	Milliron	98.2	97.8	97.0	97.6	84	78
	Bishop	98.8	98.4	98.3	97.8	78	72
	Ferguson	98.6	98.4	97.2	97.2	72	70

the tunnel varied from 350 to 400 ft., depending upon the number of persons in the tunnel. With the two subjects remaining in the tunnel all the time the velocity of the air was kept at 350 ft. per min. However, twice every 5 minutes two of the observers entered the tunnel to take various readings, raising thus temporarily the velocity of air to 400 ft. per min.

The work performed consisted in raising a known weight attached to a rope, the latter passing over a pulley and provided with a suitable handle at the other end. The weights were so adjusted that a force of 40 lb. was required to overcome friction of rope and weight guides, and raise the weight at a uniform rate. The distance through which the weights were raised was fixed to 5 ft., so that each pull represented 200 ft-lb. of work done. Each subject was required to raise the weight 75 times in 5 min. He then rested for the next 5 min., and repeated the work alternately thereafter until compelled to leave the chamber either because of fatigue or because of the severity of exposure. At the lower temperatures the subjects

were not required to remain in the test chambers longer than 3 hours. During the tests the subjects were permitted to drink ice water in any quantity and at any

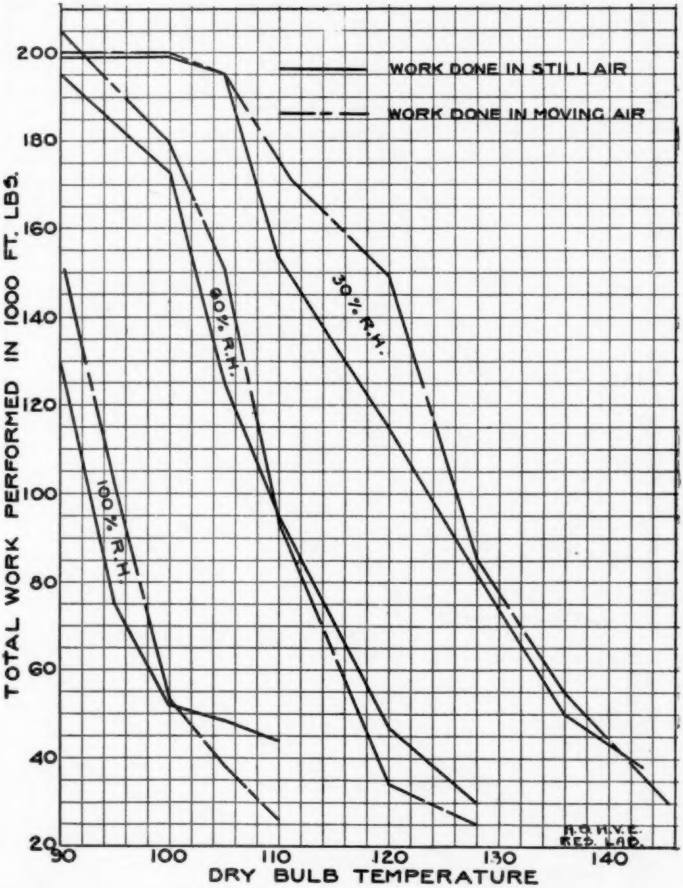


FIG. 1. AVERAGE OF TOTAL WORK THE SUBJECTS WERE CAPABLE OF PRODUCING IN THE STILL AND MOVING AIR EXPERIMENTS

time they wished. The clothing worn by the subjects during the experiments consisted of light-weight work trousers and shoes, very similar to that worn in the previous experiments and in the development of the basic scale of effective temperatures.

Observations of bodily reactions to the combined effect of work and temperature were made at the beginning and at the end of the 5-min. working periods. It was found necessary, especially in this series of tests to increase the number of

observers to four. One of them devoted his whole time in recording the rectal, mouth and surfaces temperatures of the subjects. Two observers were required

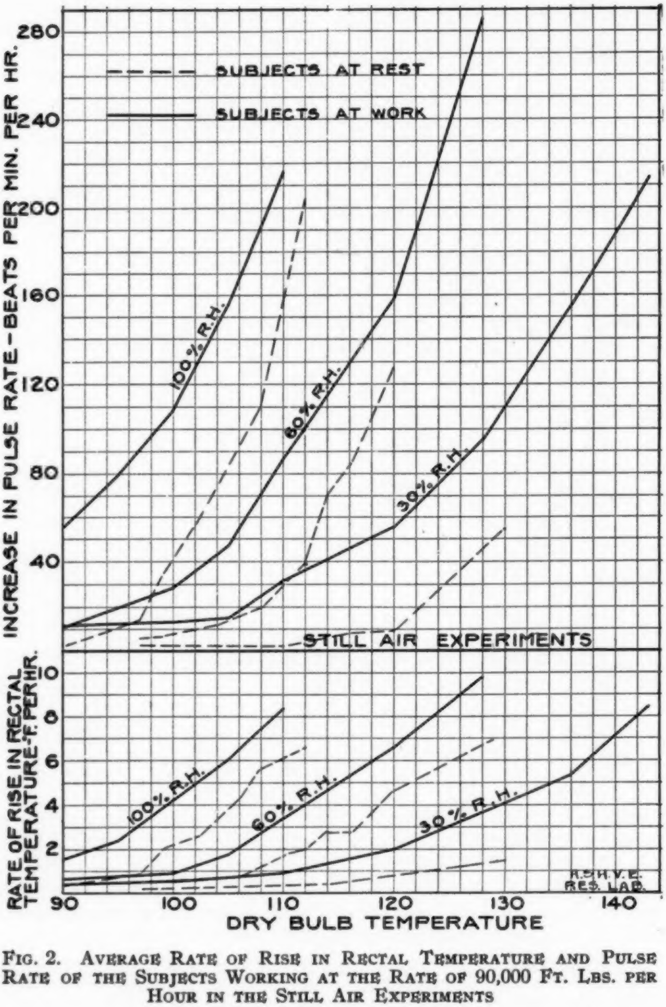


FIG. 2. AVERAGE RATE OF RISE IN RECTAL TEMPERATURE AND PULSE RATE OF THE SUBJECTS WORKING AT THE RATE OF 90,000 FT. LBS. PER HOUR IN THE STILL AIR EXPERIMENTS

for recording pulse rate and checking up on the work done, while the fourth maintained the temperature conditions and occasionally took samples of urine and sweat.

The systolic and diastolic pressures of the subjects were recorded immedi-

ately before and after the tests, and loss in body weight was determined from the weight before and after exposure. Great care has been taken to overcome the difficulties encountered in previous tests, consequently the experimental errors in determining the various measurable quantities have been reduced to a minimum.

Pulse Rate and Temperature During Test

The pulse rate and temperature of one subject is shown in Table 2 before, during, and after tests of 110 deg. dry bulb with relative humidities of 30, 60 and 100 per cent respectively. The letter B indicates that the reading was determined before the subject raised the weight and A indicates the reading immediately afterwards. These data were recorded for all subjects. Table 2 follows:

TABLE 2. DRY-BULB TEMPERATURE 110 DEG. FAHR.

Relative Humidity	30%			60%			100%		
Time		Temperature			Temperature			Temperature	
	Pulse	Rectal	Mouth	Pulse	Rectal	Mouth	Pulse	Rectal	Mouth
P.M.									
1:00	84	99.1	97.7	84	99.0	97.0	78	99.2	96.8
1:30	78	98.7	97.8	78	98.7	97.5	72	98.0	97.0
2:00	72	98.4	97.8	72	98.3	98.0	72	97.6	97.6
2:30	72	98.3	97.6	80	98.0	98.0	72	97.6	97.4
3:00	78	98.4	97.3	84	98.2	97.3	68	97.8	96.4
		Test Begun			Test Begun			Test Begun	
3:05	72B			84B			84B		
3:10	88A	98.6	98.1	126A	98.7	98.9	124A	99.3	101.7
3:15	92B			102B			124B		
3:20	102A	98.7	98.1	146A	99.2	99.7	188A	101.5	104.5
3:25	96B			128B			158B		
3:30	110A	99.0	98.4	136A	99.9	100.4	180A	102.0	104.6
3:35	106B			132B			Test Ended 3:28		
3:40	112A	99.2	98.4	144A	100.5	101.0			
3:45	106B			148B					
3:50	114A	99.4	98.6	154A	101.2	101.6			
3:55	104B			140B					
4:00	126A	99.6	98.6	172A	101.9	101.7	96	100.8	99.0
4:05	106B			152B					
4:10	120A	99.8	98.8	160A					
4:15	110B			Test Ended					
4:20	128A	100.1	98.8						
4:25	Discontinued to repair machine								
4:30				120	101.8	100			
4:35	108B								
4:40	128A	100.1	99.0						
4:45	114B						92	99.8	98.8
4:50	136A	100.4	99.2						
4:55	122B								
5:00	146A	100.4	99.4	90	100.3	98.4	84	99.4	98.4
5:05	139B								
5:10	140A	100.6	99.0						
	56 pulls								
5:15	134			84	99.8	98.2	74	99.0	98.2
	Test ended								
5:30	96	100.5	98.4						
5:45	92	100.4	98.6						
6:00	82	100.0	98.2						

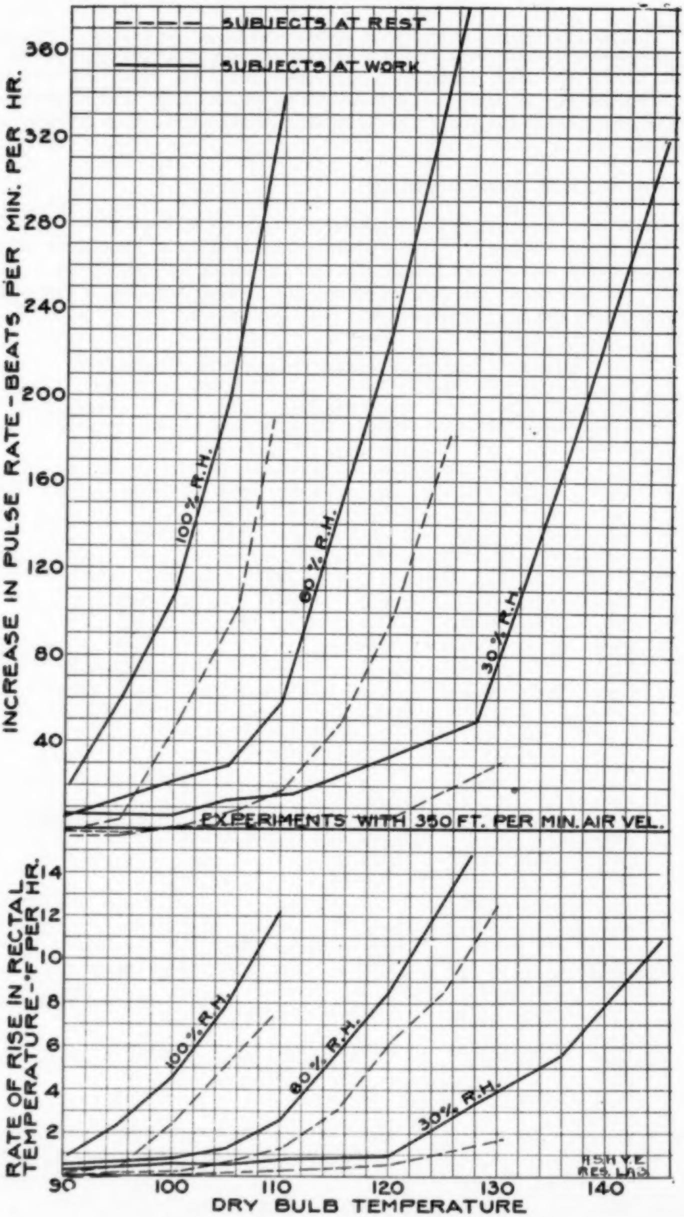


FIG. 3. AVERAGE RATE OF RISE IN RECTAL TEMPERATURE AND PULSE RATE OF THE SUBJECTS WORKING AT THE RATE OF 90,000 FT. LBS. PER HOUR IN THE MOVING AIR EXPERIMENTS

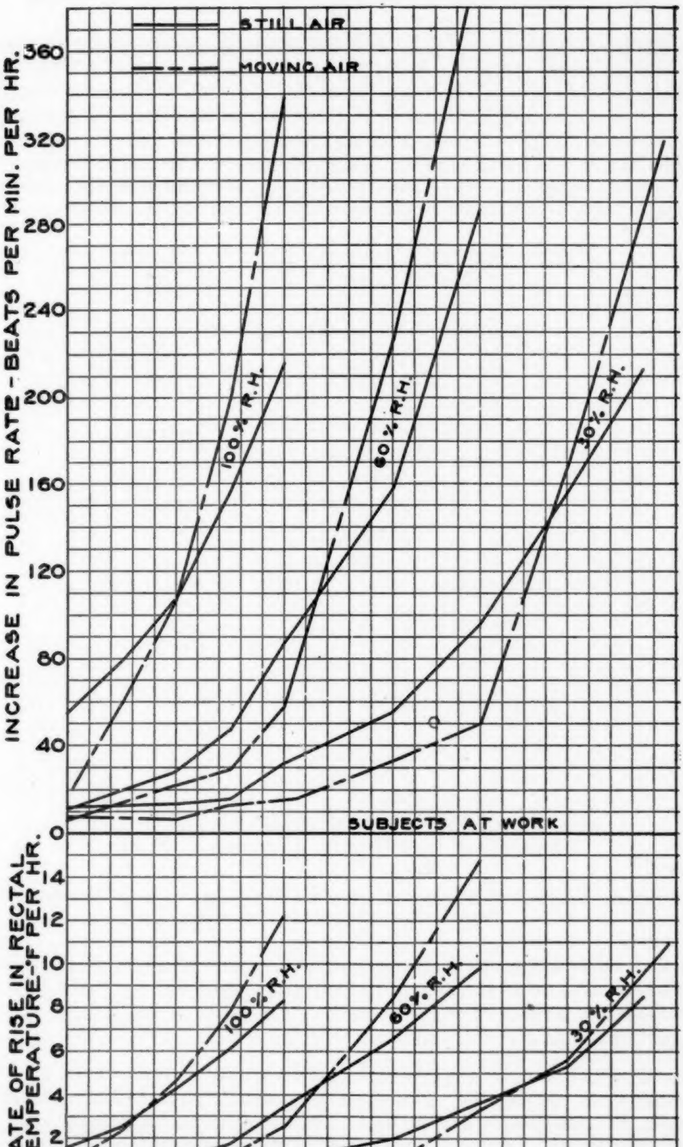


FIG. 4. COMPARISON OF PHYSIOLOGICAL REACTIONS ATTAINED WITH AND WITHOUT AIR MOVEMENT

Preliminary observations of pulse and temperature changes during work disclosed that the pulse rate dropped considerably during the resting period. The rectal temperature did not follow the same course, but with the fixed rate at which the work was done it continued increasing at a rate depending upon the atmospheric condition. This is probably due to the high heat capacity of the body and the time required for its various parts to attain the same temperature. The first observations on body temperature were made by means of a potentiometer, the rectal thermocouple being kept inserted in place through the test period. However, it was

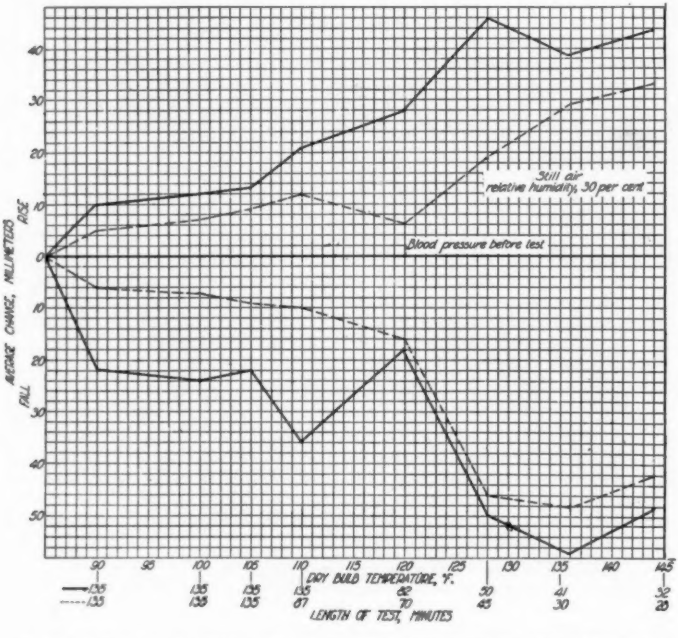


FIG. 5. CHANGE IN BLOOD PRESSURE ON EXPOSURE TO HIGH TEMPERATURES WITH 30 PER CENT HUMIDITY

found advisable later to use the 1-min. clinical thermometer inside the chambers in connection with a cold bath, and record the rectal temperature 1 or 2 min. after the 5-min. working period.

Results of Experiments

The data and results of the experiments are presented in Tables 3 and 4 for still and moving air, respectively. Individual changes in the physiological reactions are given in the usual manner in order that a direct comparison can be made with previous experiments. Column 7 gives the gross working period (including alternate periods of rest) from the beginning of the first pull to the last pull at the end of the test, and Column 8 shows the total number of times the weight was raised during this working period. This latter quantity is multiplied by 200 in Column 9 to

obtain the total work done in foot-pounds in each test. The initial temperature and pulse rate of the subjects are given in Columns 10 and 13. Whenever there was a change in these measurements from the time of entering the test chamber to the time the work was started the values are shown on the left of the column.

The rate of rise in body temperature is computed in Column 12 separately from each subject, by dividing the total rise in degrees fahrenheit by the working period in hours. In the majority of cases the average rate of rise in body temperature for each test represents the average of the four subjects employed in the experiments.

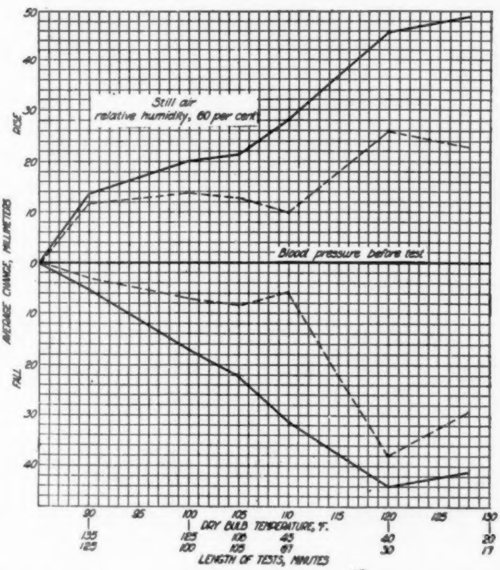


FIG. 6. CHANGE IN BLOOD PRESSURE ON EXPOSURE TO HIGH TEMPERATURES WITH 60 PER CENT HUMIDITY

It has been necessary to carry out a few experiments with only three subjects, but never with less than three.

In computing the rate of increase in the pulse frequency the average of the pulse rates before and after the last 5-min. working period is taken as the average pulse at the end of test. From this the initial pulse is subtracted to obtain the increase in the pulse rate during the working period, which divided by the average time in Column 17 gives the rate of increase in the pulse frequency shown in Column 18 for each subject separately. The body weight, loss, and rate of loss are given in the last three columns.

Discussion of Results

One of the most significant facts to be first brought out is the effect of atmospheric conditions upon the amount of work the subjects were capable of performing. In Fig. 1 the average number of foot-pounds of work performed in the experiments is

TABLE 3. RESULTS OF EXPERIMENTS WITH STILL AIR

Test and Date 1924	Test Conditions	Subjects	Time Work Entered Chamber	Left Chamber	Period Exp'd (Hr.)	Work Period (Hr.)	Total No. Pulls	Total Work Performed (Ft. Lb.)
30 D		McChesney				2.08	975	195,000
5—7	D. B. 90.0	Milliron		5:15	2.25	2.08	975	195,000
	W. B. 67.4		3:00					
19 D	R. H. 30%	Bishop		5:15	2.25	2.08	975	195,000
4—2		Ferguson		5:20	2.33	2.25	1050	210,000
								Av. 198,800
32 D		McChesney		5:10	2.17	2.08	975	195,000
5—12	D. B. 100.0	Milliron		5:15	2.25	2.08	975	195,000
	W. B. 74.4		3:00					
20 D	R. H. 30%							
4—4		Ferguson		5:15	2.25	2.25	1050	210,000
								Av. 198,800
39 D		McChesney				2.08	975	195,000
5—28	D. B. 105.0	Milliron		5:15	2.25	2.08	975	195,000
	W. B. 78.0		3:00					
16 D	R. H. 30%	Bishop		5:18	2.30	2.08	975	195,000
3—26		Ferguson		5:16	2.26	2.08	975	195,000
								Av. 195,000
40 D	D. B. 105.0	McChesney		4:40	1.67	1.42	675	135,000
6—4	W. B. 87.0	Milliron	3:00	4:30	1.50	1.08	525	105,000
	R. H. 49%							
								Av. 120,000
28 D		McChesney				1.92	900	180,000
4—28	D. B. 110.0	Milliron		5:15	2.25	1.90	881	176,200
	W. B. 82.1		3:00					
6 D	R. H. 30%	Bishop		4:28	1.47	1.25	600	120 000
3—3		Ferguson		4:35	1.59	1.42	675	135,000
								Av. 152,800
29 D		McChesney		3:55	1.42	1.42	675	135,000
5—5	D. B. 120.0	Milliron	2:30	4:05	1.59	1.42	675	135,000
	W. B. 88.9							
5 D	R. H. 30%	Bishop		4:18	1.30	0.75	375	75,000
2—29		Ferguson	3:00	4:23	1.38	0.75	375	75,000
								Av. 105,000
25 D		McChesney		3:47	0.78	0.75	375	75,000
4—21	D. B. 128.0	Milliron		3:50	0.83	0.75	375	75,000
	W. B. 95.0		3:00					
9 D	R. H. 30%	Bishop		3:45	0.75	0.73	355	71,000
3—10		Ferguson		4:14	1.23	1.08	525	105,000
								Av. 81,500
38 D		McChesney				0.70	330	66,000
5—26	D. B. 136.0	Milliron		3:42	0.70	0.58	300	60,000
	W. B. 100.7		3:00					
14 D	R. H. 30%	Bishop		3:26	0.45	0.25	150	30,000
3—21		Ferguson		3:32	0.54	0.42	225	45,000
								Av. 50,300
36 D		McChesney		2:57	0.45	0.42	225	45,000
5—21	D. B. 143.0	Milliron	2:30	3:02	0.54	0.42	225	45,000
	W. B. 107.5							
8 D	R. H. 31%	Bishop		3:16	0.27	0.25	150	30,000
3—7		Ferguson	3:00	3:24	0.40	0.25	150	30,000
								Av. 37,500
35 D		McChesney				2.08	975	195,000
5—19	D. B. 90.0	Milliron		5:15	2.25	2.08	975	195,000
	W. B. 78.3		3:00					
18 D	R. H. 60%	Bishop		5:05	2.08	1.92	900	180,000
3—31		Ferguson		5:20	2.33	2.25	1050	210,000
								Av. 195,000

Table 3. (Continued)

Rectal Temperature: Initial Primary Room	Rectal Temperature: Initial Test Room	Total Rise °F.	Rate Rise per °F. Hr).	Pulse Rate (Beats per Min.): Initial Primary Rm.	Initial Test Rm.	Last Work Period Final Before	Last Work Period Final After	Av. Pulse End of Test	Av. Increase	Av. Time	Av. Rate Increase	Body Wt. (lb.) Initial	Loss	Rate of Loss Lb. per Hr.
99.3		1.1	0.53	80		88	112	100	20	2.04	10	144.19	1.44	0.64
98.3	98.3	1.5	0.72	78	80	104	128	116	36	2.04	18	131.00	1.00	0.44
99.1	99.1	0.9	0.43	78	80	100	104	102	22	2.04	11	122.63	1.42	0.63
98.6		0.4	0.18	74		90	92	91	17	2.21	8	142.44	1.63	0.70
		Av.	0.47							Av.	12		Av.	0.60
99.2		0.7	0.34	84		84	118	101	17	2.04	8	145.00	2.25	1.04
98.1	98.1	1.5	0.72	72	76	100	136	118	42	2.04	21	129.87	2.37	1.05
98.6		0.9	0.40	70		84	104	94	24	2.21	11	140.00	1.13	0.50
		Av.	0.49							Av.	13		Av.	0.86
98.6	98.6	1.1	0.53	60	80	88	128	108	28	2.04	14	144.37	2.50	1.11
98.2		2.2	1.06	72		120	132	126	54	2.04	26	129.19	2.69	1.20
98.7	98.4	1.2	0.57	80	80	88	108	98	18	2.04	9	121.00	2.00	0.87
98.4		0.3	0.14	70		80	104	92	22	2.04	11	143.63	2.75	1.22
		Av.	0.58							Av	15		Av	1.10
99.0		2.0	1.41	80		132	141	137	57	1.38	41	143.75	3.25	1.98
97.8	97.8	3.0	2.78	78	92	132	156	144	52	1.04	50	129.31	2.37	1.55
		Av.	2.10							Av.	45		Av.	1.77
98.7		1.6	0.83	84		108	120	114	30	1.88	16	143.44	3.37	1.50
98.4	98.4	2.2	1.16	78	72	128	140	134	62	1.87	33	129.75	2.19	0.97
99.0	98.8	1.3	1.04	68	78	110	138	124	46	1.21	38	121.13	1.37	0.93
98.8		1.2	0.85	78		124	140	132	54	1.38	39	143.25	2.25	1.41
		Av.	0.97							Av.	32		Av.	1.20
98.9		2.3	1.62	84		130	146	138	54	1.38	39	145.31	3.81	2.68
98.1	98.2	3.6	2.53	80	86	128	146	137	51	1.38	37	130.00	2.44	1.53
99.1	98.8	1.9	2.53	70	76	118	146	132	56	0.71	79	121.25	2.86	2.20
98.6		1.0	1.33	70			120		50	0.75	67	144.94	2.00	1.45
		Av.	2.00							Av.	56		Av.	1.97
99.0		2.3	3.07	96		130	172	151	55	0.71	78	145.87	2.50	3.20
98.4	98.4	3.7	4.93	84	78	146	170	158	80	0.71	112	131.81	1.63	1.97
98.9		2.7	3.70	78		132	172	152	74	0.70	106	121.50	1.50	2.00
98.6	98.6	3.4	3.15	74	78	150	185	168	90	1.04	87	143.31	3.94	3.20
		Av.	3.71							Av.	96		Av.	2.59
99.3		3.1	4.43	80		156	180	168	88	0.68	129	142.75	2.50	3.57
98.2	98.2	3.6	6.20	84	102	156	188	172	70	0.54	129	130.00	2.35	3.36
99.6	99.6	1.2	4.80	86	108	142	160	151	43	0.21	205	120.75	3.00	6.67
98.6		2.5	5.94	72		116	150	133	60	0.38	158	140.75	2.44	4.52
		Av.	5.34							Av.	155		Av.	4.53
99.7		2.1	5.00	76		148	176	162	86	0.38	227	143.75	1.50	3.33
98.0	98.4	4.2	10.00	72	100	148	168	158	58	0.38	153	129.75	1.37	2.54
99.1		2.8	11.20	68		132	152	142	74	0.21	352	122.13	2.03	7.53
98.6	99.8	1.9	7.60	70	110	122	148	135	25	0.21	119	141.37	1.37	3.42
		Av.	8.45							Av.	213		Av.	4.21
99.4		0.8	0.38	84		100	104	102	18	2.04	9	143.63	1.25	0.56
98.0	98.0	1.6	0.77	72	74	102	108	105	31	2.04	15	128.00	1.63	0.72
98.0	98.0	1.8	0.94	78	72	90	102	96	24	1.88	13	121.31	1.13	0.54
98.6	98.6	0.6	0.27	76	72	78	90	84	12	2.21	5	143.37	1.81	0.78
		Av.	0.59							Av.	11		Av.	0.65

TABLE 3. (CONCLUDED)

Rectal Temperature: Initial Primary Room	Initial Test Room	Total Rise °F.	Rate Rise per °F. Hr.	Pulse Rate (Beats Per Minute): Initial Primary Rm.	Initial Test Rm.	Last Work Period Final Before	Last Work Period Final After	Av. Pulse End of Test	Av. Increase	Av. Time	Av. Rate Increase	Body Wt. (lb.) Initial	Loss	Rate of Loss Lb. per Hr.
98.8		1.3	0.63	80		96	124	110	30	2.04	15	145.63	3.13	1.45
98.4	98.4	2.4	1.25	72	80	156	160	158	78	1.88	42	127.87	2.25	1.08
99.0		1.4	0.89	70		120	130	125	55	1.54	36	121.25	2.87	1.70
98.4	98.4	1.2	0.69	74	78	104	120	112	34	1.71	20	142.19	2.63	1.32
		Av.	0.87								Av. 28			Av. 1.39
99.2		1.6	1.74	76		120	140	130	54	0.88	61	142.13	2.75	2.75
98.4	98.4	3.7	2.34	80	86	146	162	154	68	1.54	44	129.50	2.75	1.54
98.6		1.6	1.13	72		108	136	122	50	1.38	36	142.37	3.13	1.69
		Av.	1.74								Av. 47			Av. 1.99
99.2		2.8	2.64	86		156	176	166	80	1.03	78	146.75	3.62	3.23
98.2	98.5	3.8	3.52	84	84	152	160	156	72	1.04	69	130.69	1.81	1.53
9.4		2.8	3.84	72		132	168	150	78	0.70	111	121.25	2.13	2.66
98.0	98.2	3.9	3.61	76	78	162	180	171	93	1.04	89	141.19	2.81	2.38
		Av.	3.40								Av. 87			Av. 2.45
99.0		1.2	4.80	84		120	132	126	42	0.21	200	146.63	1.13	3.76
98.2	98.9	3.8	7.04	72	92	156	189	173	81	0.52	156	131.13	1.19	1.92
99.6	99.8	2.6	6.18	72	96	138	162	150	54	0.38	142	122.00	1.75	3.50
98.6		4.8	8.27	76		138	160	149	73	0.54	135	143.75	2.75	4.58
		Av.	6.57								Av. 158			Av. 3.44
99.3		2.2	8.80	76		128	180	154	78	0.21	372	145.44	1.44	5.76
98.4	98.4	2.3	9.20	84	108	148	190	169	61	0.21	291	129.31	1.37	4.15
99.0		2.4	10.00	76		120	150	135	59	0.21	282	120.63	1.00	3.57
98.6	98.7	2.8	11.20	70	96	120	156	138	42	0.21	200	141.31	1.69	5.28
		Av.	9.80								Av. 286			Av. 4.69
98.2		3.4	1.63	76		132	180	156	80	2.04	39	132.50	1.87	0.83
98.4	98.4	1.4	1.52	76	80		146		66	0.92	72	119.00	2.37	1.90
98.6	98.6	1.6	1.48	74	78		136		58	1.08	54	143.56	2.25	1.39
		Av.	1.54								Av. 55			Av. 1.37
97.8	97.9	3.1	2.87	72	78	150	168	159	81	1.04	79	130.06	2.31	1.85
99.1	99.2	1.9	2.29	80	90	132	172	152	62	0.79	79	120.50	1.25	1.11
98.8	98.8	1.6	2.13	80	84	120	262	141	57	0.71	80	143.63	1.63	1.63
		Av.	2.43								Av. 79			Av. 1.53
98.9	99.1	1.2	2.86	80	102	120	158	139	37	0.38	97	146.13	1.75	3.07
98.4		4.2	5.60	84		134	168	151	67	0.71	194	131.69	1.56	2.00
98.9	99.1	1.5	4.17	76	88	126	144	135	47	0.35	134	120.75	2.00	2.56
		Av.	4.21								Av. 108			Av. 2.54
99.4		1.61	3.80	96		138	154	146	50	0.38	132	147.50	2.87	6.10
98.2	98.5	4.31	7.42	72	90	156	192	174	84	0.54	155	133.75	1.71	2.55
99.0	99.0	1.7	6.80	74	94	102	156	129	35	0.21	167	121.56	1.16	2.76
99.0		3.5	6.25	72		130	194	162	90	0.53	170	145.13	2.50	4.24
		Av.	6.07								Av. 156			Av. 3.91
98.8		2.8	7.00	88		144	154	149	61	0.37	165	146.75	1.06	2.26
97.8	98.0	4.0	9.52	68	84	158	188	173	89	0.38	234	132.81	2.40	4.90
98.6		3.6	8.57	80		160	189	175	95	0.38	250	143.87	1.31	3.12
		Av.	8.36								Av. 216			Av. 3.43

TABLE 4. RESULTS OF EXPERIMENTS WITH 350 FT. PER MIN. AIR VELOCITY

Test and Date 1924	Test Conditions	Subjects	Time Work Entered Chamber	Time Work Left Chamber	Period of Exp'd (Hrs.)	Work Period (Hrs.)	Total No. Pulls	Total Work Performed (Ft. Lbs.)
19 D		McChesney		5:20	2.33	2.25	1050	210,000
4—2	D. B. 90.0							
	W. B. 67.3		3:00					
30 D	R. H. 30%	Bishop				2.08	975	195,000
5—7		Ferguson		5:15	2.25	2.08	975	195,000
								Av. 200,000
20 D		McChesney		5:15	2.25	2.25	1050	210,000
4—4	D. B. 100.0							
	W. B. 74.4		3:00					
32 D	R. H. 30%	Bishop		5:10	2.16	2.08	975	195,000
5—12		Ferguson		5:15	2.25	2.08	975	195,000
								Av. 200,000
16 D		McChesney		5:15	2.25	2.08	975	195,000
3—26	D. B. 105.0	Milliron		5:18	2.30	2.08	975	195,000
	W. B. 78.0		3:00					
39 D	R. H. 30%	Bishop				2.08	975	195,000
5—28		Ferguson		5:15	2.25	2.08	975	195,000
								Av. 195,000
40 D	D. B. 105.0							
6—4	W. B. 87.0	Bishop		4:40	1.67	1.58	750	150,000
	R. H. 49%	Ferguson	3:10	4:45	1.75	1.58	750	150,000
								Av. 150,000
6 D		McChesney		4:33	1.55	1.25	600	120,000
3—26	D. B. 111.0	Milliron		5:10	2.16	1.75	825	165,000
	W. B. 82.2		3:00					
28 D	R. H. 30%	Bishop		5:15	2.25	2.08	975	195,000
4—28		Ferguson		5:18	2.30	2.25	1050	210,000
								Av. 172,300
5 D		McChesney	3:00	3:55	0.92	0.58	300	60,000
2—29	D. B. 120.0							
	W. B. 89.0							
29 D	R. H. 30%	Bishop		4:45	2.25	2.07	965	193,000
5—5		Ferguson	2:30	4:40	2.16	2.08	975	195,000
								Av. 149,300
9 D		McChesney		3:45	0.75	0.73	360	72,000
3—10	D. B. 128.0	Milliron		3:54	0.90	0.75	375	75,000
	W. B. 95.0		3:00					
23 D	R. H. 30%	Bishop		4:04	1.07	0.92	450	90,000
4—21		Ferguson		4:09	1.15	1.08	525	105,000
								Av. 85,500
14 D		McChesney		3:31	0.52	0.42	225	45,000
3—21	D. B. 136.0	Milliron		3:26	0.44	0.42	225	45,000
	W. B. 100.7		3:00					
38 D	R. H. 30%							
5—26		Ferguson		3:45	0.75	0.75	375	75,000
								Av. 55,000
3 D		McChesney		3:25	0.42	0.25	150	30,000
3—7	D. B. 145.4	Milliron	3:00	3:20	0.33	0.25	150	30,000
	W. B. 107.1							
36 D	R. H. 30%	Bishop		2:46	0.27	0.25	150	30,000
5—21		Ferguson	2:30	2:50	0.33	0.25	150	30,000
								Av. 30,000
18 D		McChesney		5:15	2.25	2.25	1050	210,000
3—31	D. B. 90.0							
	W. B. 78.3		3:00					
35 D	R. H. 60%	Bishop				2.08	975	195,000
5—19		Ferguson		5:15	2.25	2.25	1050	210,000
								Av. 205,000

TABLE 4. (CONTINUED)

Rectal Temperature: Initial Primary Room	Rectal Temperature: Initial Test Room	Total Rise °F.	Rate Rise °F. per Hr.	Pulse Rate (Beats per Min.): Initial Primary Rm.	Initial Test Rm.	Last Work Period Final Before	Last Work Period Final After	Av. Pulse End Test	Av. Increase	Av. Time	Av. Rate Increase	Body Wt. (lb.) Initial	Loss	Rate Loss Lb. per Hr.
99.3		0.6	0.27	84		94	114	104	20	2.21	9	148.87	1.87	0.80
99.1	99.1	0.5	0.24	72	78	88	100	94	16	2.04	8	120.37	1.87	0.83
99.0		0.4	0.19	84		88	108	98	14	2.04	7	141.63	1.63	0.72
		Av.	0.23							Av.	8		Av.	0.78
98.4		1.5	0.67	84		98	100	99	15	2.21	7	148.00	3.87	1.72
98.4	98.6	1.5	0.72	70		84	94	89	19	2.04	9	120.63	3.13	1.45
99.3		0.5	0.24	74	78	78	98	88	10	2.04	5	142.37	2.75	1.22
		Av.	0.54							Av.	7		Av.	1.46
98.9		1.1	0.53	80		102	128	115	35	2.04	17	148.63	3.50	1.56
97.9	98.4	1.3	0.62	72	78	108	132	120	42	2.04	21	132.19	3.69	1.60
99.0	99.0	1.1	0.53	80	84	84	102	93	9	2.04	4	121.00	2.25	1.00
98.6		0.5	0.24	70		84	102	93	23	2.04	11	142.00	2.87	1.28
		Av.	0.48							Av.	13		Av.	1.36
98.4		2.3	1.46	72		102	142	122	50	1.54	32	120.87	2.75	1.65
98.4	98.4	0.8	0.51	70	72	96	108	102	30	1.54	20	140.50	2.00	1.14
		Av.	0.99							Av.	26		Av.	1.40
98.7		1.4	1.12	90		98	136	117	27	1.21	22	143.69	2.94	1.90
98.4	98.4	1.6	0.92	72	92	114	126	120	28	1.71	16	130.75	3.13	1.45
98.0	98.4	1.6	0.77	70	78	90	126	108	30	2.04	15	122.37	2.87	1.28
99.2		0.8	0.36	78		96	128	112	34	2.21	15	141.87	4.31	1.87
		Av.	0.79							Av.	17		Av.	1.63
99.1	99.1	0.9	1.55	76	92	100	150	125	33	0.54	61	147.69	1.31	1.42
99.1	99.1	1.4	0.68	76	88	100	120	110	22	2.04	11	119.37	3.13	1.39
99.6		0.7	0.34	78		114	148	131	53	2.04	26	140.75	4.50	2.08
		Av.	0.86							Av.	33		Av.	1.63
98.4		2.0	2.74	84		124	146	135	51	0.70	73	147.44	2.69	3.59
98.0	98.0	3.7	4.93	90	100	128	130	129	29	0.71	41	131.50	2.13	2.37
98.6	98.6	2.8	3.04	76	90	120	150	125	35	0.88	40	122.13	1.50	1.40
98.8		2.8	2.58	78		120	150	125	47	1.04	45	142.31	3.19	2.77
		Av.	3.32							Av.	50		Av.	2.53
99.4	99.4	2.2	5.23	84	96	156	164	160	64	0.38	168	145.75	2.63	5.09
98.2		2.3	5.49	78		138	180	159	81	0.38	213	129.63	1.87	4.25
98.6		4.4	5.87	74		150	168	159	85	0.71	120	141.50	3.19	4.26
		Av.	5.53							Av.	167		Av.	4.52
99.4		2.4	9.60	88		144	162	153	65	0.21	309	145.63	1.44	3.43
98.4	98.4	2.6	10.40	84	114	132	192	162	48	0.21	228	132.81	1.87	5.66
99.2		3.0	12.00	74		156	174	165	91	0.21	433	120.50	1.25	4.63
99.0	99.0	2.9	11.60	70	102	156	174	165	63	0.21	300	139.63	2.25	6.82
		Av.	10.90							Av.	318		Av.	5.14
99.2		0.8	0.36	92		92	100	96	4	2.21	2	146.50	1.87	0.83
98.0	98.4	1.4	0.67	70	72	84	102	93	21	2.04	10	121.25	1.37	0.61
99.0		0.8	0.36	78		84	102	93	15	2.21	7	141.87	1.81	0.80
		Av.	0.46							Av.	6		Av.	0.75

Table 4. Results of Experiments with 350 Ft. per Min. Air Velocity

Test Date 1924	Test Conditions	Subjects	Time Work Period Entered Chamber	Left Chamber	Exp'd (Hrs.)	Work Period (Hrs.)	Total No. of Pulls	Work Performed (Ft. Lbs.)
10 D		McChesney		5:00	2.00	1.92	900	180,000
3—12	D. B. 100.0	Milliron		5:03	2.05	1.92	900	180,000
	W. B. 87.0		3:00					
33 D	R. H. 60%	Bishop		5:00	2.00	1.92	900	180,000
5—14		Ferguson		5:05	2.08	1.92	900	180,000
								Av. 180,000
13 D		McChesney		4:33	1.55	1.25	600	120,000
3—19	D. B. 105.0	Milliron	3:00	4:42	1.70	1.36	623	124,600
	W. B. 91.5							
37 D	R. H. 60%	Bishop		4:30	2.08	1.92	900	180,000
5—23		Ferguson	2:30	4:35	2.08	1.92	900	180,000
								Av. 151,200
11 D		McChesney		4:06	1.10	0.92	450	90,000
3—14	D. B. 110.0	Milliron		3:49	0.82	0.58	300	60,000
	W. B. 96.0		3:00					
22 D	R. H. 60%	Bishop		4:10	1.16	1.08	525	105,000
4—9		Ferguson		4:22	1.37	1.25	600	120,000
								Av. 93,800
12 D		McChesney		3:29	0.49	0.42	225	45,000
3—17	D. B. 120.0	Milliron		3:26	0.44	0.25	150	30,000
	W. B. 104.8		3:00					
26 D	R. H. 60%	Bishop		3:18	0.30	0.25	150	30,000
4—23		Ferguson		3:24	0.40	0.25	150	30,000
								Av. 33,800
17 D		McChesney		2:15	0.25	0.22	125	25,000
3—28	D. B. 128.0	Milliron	2:00	2:16	0.27	0.20	100	20,000
	W. B. 112.0							*
31 D	R. H. 60%							
5—9		Ferguson	2:30	2:45	0.25	0.25	150	30,000
								Av. 25,000
2 D		McChesney		4:20	1.33	1.06	500	100,000
2—11	D. B. 90.4	Milliron		4:31	1.52	1.06	500	100,000
	W. B. 90.4		3:00					
27 D	R. H. 100%	Bishop		5:15	2.25	2.08	975	195,000
4—25		Ferguson		5:20	2.33	2.25	1050	210,000
								Av. 151,200
4 D*		McChesney		4:16	1.27	1.06	500	100,000
2—27	D. B. 95.0	Milliron	3:00	4:12	1.20	1.00	420	84,000
	W. B. 95.0							
34 D	R. H. 100%	Bishop		3:40	1.16	1.08	525	105,000
5—16		Ferguson	2:30	3:55	1.42	1.25	600	120,000
								Av. 102,300
3 D		McChesney		3:37	0.62	0.36	170	34,000
2—25	D. B. 100.0	Milliron		3:43	0.72	0.53	250	50,000
	W. B. 100.0		3:00					
24 D	R. H. 100%	Bishop		3:35	0.59	0.54	260	52,000
4—14		Ferguson		3:53	0.88	0.75	375	75,000
								Av. 52,800
15 D		McChesney		3:27	0.45	0.42	225	45,000
3—24	D. B. 105.0	Milliron		3:24	0.40	0.25	150	30,000
	W. B. 105.0		3:00					
21 D	R. H. 100%	Bishop		3:22	0.37	0.25	150	30,000
4—7		Ferguson		3:27	0.45	0.42	225	45,000
								Av. 37,500
7 D		McChesney		3:18	0.30	0.22	125	25,000
3—5	D. B. 110.0	Milliron		3:20	0.33	0.22	125	25,000
	W. B. 110.0		3:00					
23 D	R. H. 100%	Bishop		3:15	0.25	0.21	115	23,000
4—11		Ferguson		3:21	0.35	0.25	150	30,000
								Av. 25,800

* Work delayed by defect in apparatus.

TABLE 4. (CONCLUDED)

Rectal Temperature: Initial Primary Room	Rectal Temperature: Initial Test Room	Total Rise °F.	Rate Rise °F. per Hr.	Pulse Rate (Beats per Min.): Initial Primary Rm.	Initial Test Rm.	Last Work Period Final Before	Last Work Period Final After	Av. Pulse End of Test	Av. Increase	Av. Time	Av. Rate Increase	Body Wt. (lb.) Initial	Loss	Rate of Loss Lb. per Hr.
99.2		1.7	0.88	80		109	156	133	53	1.88	28	146.13	3.13	1.57
98.0	98.3	2.0	1.04	76	86	120	140	130	44	1.88	23	132.69	3.13	1.53
98.3		1.5	0.78	70		96	118	107	37	1.88	20	122.13	2.50	1.25
99.2	99.2	0.4	0.21	76	78	96	120	108	30	1.88	16	141.63	2.44	1.17
			Av. 0.73								Av. 22			Av. 1.38
98.9		1.7	1.36	88		120	122	121	33	1.21	27	145.87	3.00	1.94
98.2	99.1	2.2	1.62	76	88	128	148	138	50	1.35	37	130.00	2.37	1.98
99.0		2.0	1.04	70		102	140	121	51	1.88	27	121.25	3.25	1.63
98.6	98.6	1.6	0.83	74	78	106	144	125	47	1.88	25	139.75	3.00	1.44
			Av. 1.21								Av. 29			Av. 1.75
99.4		3.5	3.80	92		122	158	140	48	0.88	55	146.19	3.31	3.01
..	..	...	..	84	88	120	124	122	34	0.54	63	131.37	1.37	1.67
99.6		2.3	2.13	80		138	164	151	71	1.04	68	122.13	2.75	2.37
98.8	98.8	2.4	1.92	70	78	112	150	131	53	1.21	44	140.50	2.63	1.92
			Av. 2.62								Av. 58			Av. 2.24
98.7		4.0	9.52	76		134	142	138	62	0.38	163	145.25	..	..
97.4	99.0	2.5	10.00	80	98	126	152	139	41	0.21	195	132.44	..	..
99.0		1.9	7.60	74		114	156	135	61	0.21	291	120.63	1.06	3.53
98.6	99.1	1.7	6.80	72	90	132	156	144	54	0.21	257	141.19	1.69	4.23
			Av. 8.48								Av 227			Av. 3.88
98.9		3.5	15.90	84		144	184	164	80	0.20	400	147.19	1.00	4.00
97.3	98.4	3.4	17.00	16	118	168	180	174	56	0.18	311	131.87	2.06	7.63
99.0		2.9	11.60	70		144	194	169	99	0.21	472	139.56	2.00	8.00
			Av. 14.83								Av. 394			Av. 6.54
99.1		1.6	1.51	92		100	120	110	18	1.03	17	141.25	..	..
98.7	99.2	1.2	1.13	72	90	108	140	124	34	1.03	33	130.06	..	..
99.0	99.2	0.9	0.43	80	84	96	120	108	24	2.04	12	121.25	2.00	0.96
98.8		1.4	0.62	70		96	120	108	38	2.21	17	140.37	2.00	0.88
			Av. 0.92								Av. 20			Av. 0.88
99.2		2.0	1.89	84	88	156	160	158	70	1.03	68	144.63	2.19	1.72
98.4	98.8	2.7	2.70	72	92	128	156	142	50	0.96	52	132.25	2.25	1.87
98.2		3.0	2.78	70		108	164	136	66	1.04	63	122.06	2.69	2.32
98.7	98.9	2.7	2.16	70	78	142	142	142	64	1.21	53	140.50	3.00	2.11
			Av. 2.38								Av. 59			Av. 2.01
99.1	99.4	1.7	4.73	76	94	120	174	147	53	0.34	156	144.75	2.37	3.82
98.4		2.7	5.08	78	110	116	140	128	18	0.51	35	130.73	2.13	2.97
99.1		2.6	4.82	80		138	150	144	64	0.52	123	121.50	1.50	2.54
98.7	99.0	2.9	3.87	70	78	142	184	163	85	0.71	119	142.00	2.25	2.56
			Av. 4.63								Av. 108			Av. 2.97
99.3		2.5	5.95	92		126	156	141	49	0.38	129	150.00	1.52	3.38
98.3	99.0	2.7	10.80	76	116	142	150	146	30	0.21	143	131.19	1.56	3.90
99.5	99.5	1.6	6.40	72	90	136	156	146	56	0.21	267	120.63	1.13	3.05
98.8		3.5	8.33	72		146	192	169	97	0.38	256	143.63	2.50	5.55
			Av. 7.87								Av. 199			Av. 3.97
98.8		2.7	12.27	88		132	162	147	59	0.20	295	145.50	1.13	3.76
98.3	98.3	2.8	12.74	72	115	146	181	164	49	0.20	245	132.13	1.19	3.60
99.3		2.6	12.38	70		132	174	153	83	0.19	437	120.37	1.63	6.25
99.1	99.9	2.8	11.20	74	96	170	180	175	79	0.21	377	141.19	2.19	6.64
			Av. 12.15								Av. 339			Av. 5.02

plotted against the dry-bulb temperature of the conditions for the three different humidities of 100, 60 and 30 per cent employed in the tests. The solid lines represent the total work done by the average subject in still air, and the broken lines the work done in moving air. The difference between the two ordinates represents the beneficial effect of wind in increasing the work output. The principal object of this figure is to show the marked decrease in the amount of work done with increases

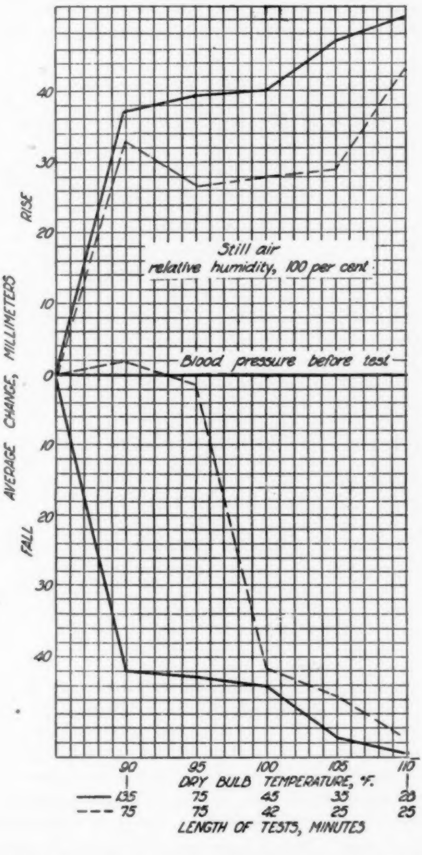


FIG. 7. CHANGE IN BLOOD PRESSURE ON EXPOSURE TO HIGH TEMPERATURES WITH 100 PER CENT HUMIDITY

in external temperature. The importance of air movement cannot be properly studied in this chart because the amount of work performed is not based on equal physiological reactions. This particular point will be considered later.

It will be observed that the subjects of the experiments were capable of performing four times more work in a temperature of 100 deg. with a relative humidity of 30 per cent than in a saturated condition of 100 deg. For the ordinary humidity of 60 per cent the subjects performed about 5 times more work in a temperature of

90 than in one of 120 deg. The rate at which the output decreases with increase in the external temperature is practically the same for all three humidities at the higher temperatures, and in the light of the experimental results the upper limit in atmospheric conditions at which work could be performed efficiently corresponds to a dry-bulb temperature of 100 deg. and a relative humidity of 30 per cent.

Figs. 2 and 3 show the average rate of rise in body temperature and average rate of increase in pulse rate plotted on the same base as Fig. 1. The general characteristics of the curves are similar to those of the *rest* experiments, as shown by the dotted lines. The effect of muscular work in increasing the bodily reactions is

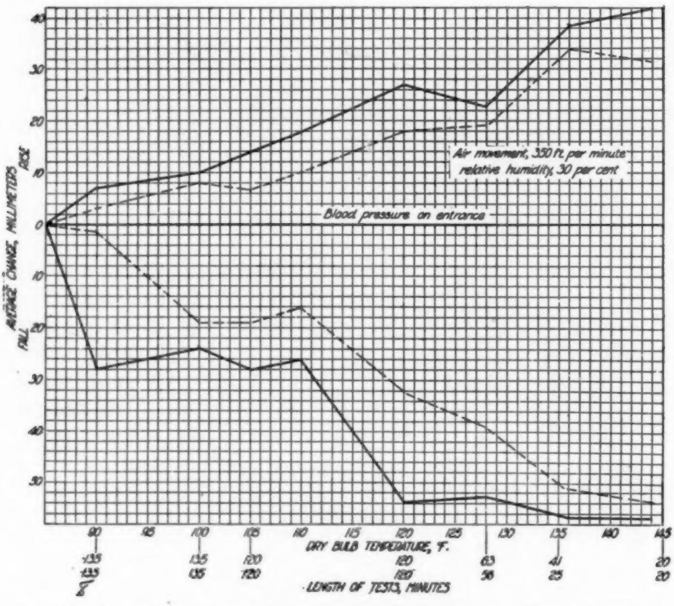


Fig. 8. Change in Blood Pressure on Exposure to High Temperatures with 30 Per Cent Humidity

clearly shown by the difference in the ordinates of the solid and dotted lines. It will be noted for example, that the average rate of rise in the rectal temperature of the subjects when working at the rate of 90,000 ft.-lb. per hour in a temperature of 110 deg. and a humidity of 60 per cent is about twice as high as the corresponding rate with the subjects at rest. This difference in the bodily reactions will naturally depend on the rate at which work is performed; the greater the rate the higher the reactions.

It has been pointed out previously that with the subjects at rest the upper limit of man's ability to compensate for atmospheric conditions lies around 90 deg. effective temperature. The present experiments show that when work is done at the rate of 90,000 ft.-lb. per hour this limit is reached at a temperature considerably lower than 90 deg. effective temperature. Unfortunately the experiments thus far

were not carried down to that temperature, but judging from the 30 per cent relative humidity curves it will be observed that a beginning rise in the principal physiological reactions takes place at about 95 deg. dry-bulb temperature which for still air corresponds to an effective temperature of about 80 deg.

To show the importance of air movement in lowering the physiological reactions of the subjects at work, Fig. 3 has been superimposed upon Fig. 2 as shown in Fig. 4. The beneficial effect of air movement when the temperature of the environment is below the temperature of the body is shown by the considerably lower reaction obtained in moving air. Especially in the pulse curves the effect of the wind is

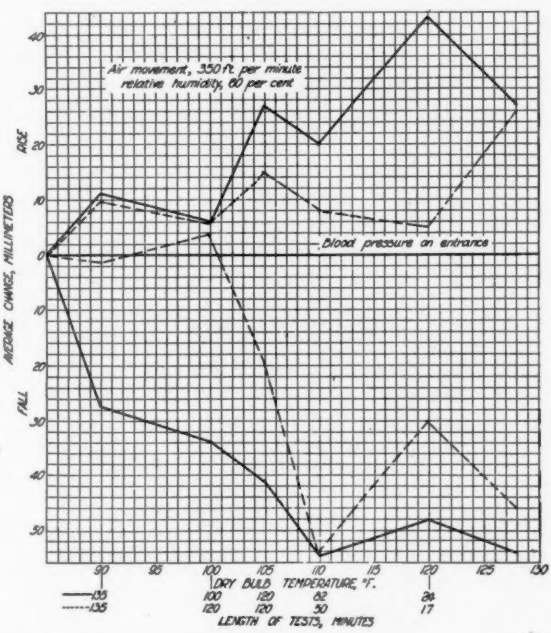


FIG. 9. CHANGE IN BLOOD PRESSURE ON EXPOSURE TO HIGH TEMPERATURES WITH 60 PER CENT HUMIDITY

well pronounced, which constitutes a further evidence in the pulse rate being a much more sensitive and dependable index of bodily reactions to atmospheric temperature and muscular work.

The heating effect of wind in temperatures above that of the body is shown by the higher reactions produced in moving air than in still air. In general it can be stated that within workable limits of temperature the air movement of 350 ft. per min. reduced the rate of increase in pulse frequency by about 45 per cent.

Blood Pressure

Figs. 5 to 10 inclusive illustrate the variations in the systolic and diastolic pressures of the blood. The heavy horizontal lines represent both pressure readings

of the subjects before exposure to the test condition. The curved lines represent the changes after exposure, the solid line giving the maximum and the dotted the minimum variation in millimeters of mercury. These are plotted against dry-bulb temperature for the three different relative humidities and for still and moving air.

These variations are very similar to those found in previous experiments. The deviations are more marked, which is to be expected on account of the subjects

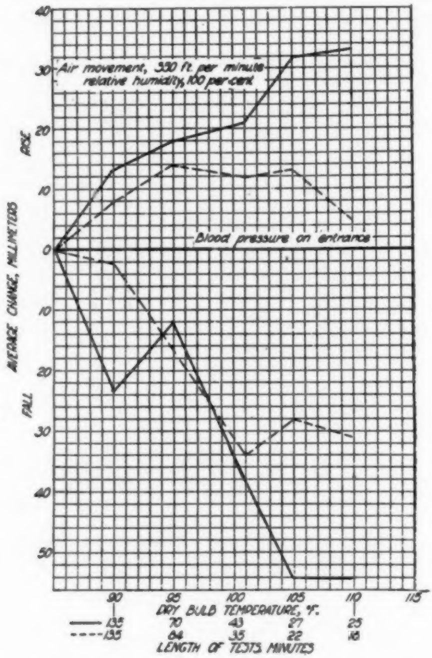


FIG. 10. CHANGE IN BLOOD PRESSURE ON EXPOSURE TO HIGH TEMPERATURES WITH 100 PER CENT HUMIDITY

working. However, seldom was the diastolic pressure less than zero, as found in the experiments at rest.

Loss of Body Weight

The average rate of loss of body weight for the still and moving air experiments is given in Fig. 11 plotted against dry-bulb temperature. These curves are not as regular as those in Figs. 2 and 3 because of the water drunk in the experiments. Although the amount of water consumed by each subject was marked in all tests and taken into consideration in calculating the rate of loss, it is obvious that the greater the water consumption during the experiments, the greater will be the perspiration and therefore the greater the loss in weight.

In regard to the effect of water on the other physiological reaction measures, the rectal temperature did not seem to have been measurably affected. The pulse rate was slightly decreased, but the mouth temperature dropped considerably. Be-

cause of this uncertainty in the mouth temperature when water was drunk, this quantity is not considered here. The chloride determinations of the sweat and urine analyses will be reported in separate articles.

Effect of Air Motion

To indicate the effect of air movement upon the ability of individuals to perform work in comparatively high temperatures, the average work done by the subjects in still and moving air at the normal relative humidity of 60 per cent has been based upon equal physiological reaction measures and shown in Fig. 12. The lower solid and broken curves give the average work done in foot-pounds per degree average rise in rectal temperature for still and moving air, respectively. In the majority of

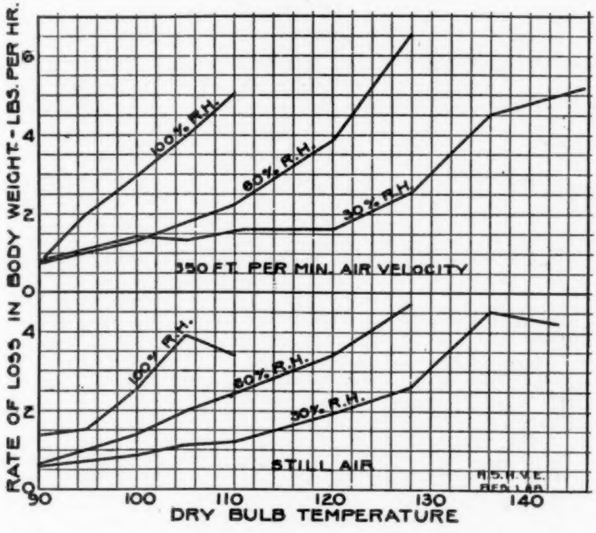


FIG. 11. AVERAGE RATE OF LOSS IN BODY WEIGHT IN THE STILL AND MOVING AIR EXPERIMENTS

these experiments a degree rise in rectal temperature was found to correspond to 25 beats increase in the pulse rate. Accordingly, the work done per 25 beats increase in the pulse rate was calculated and shown in Fig. 12 by the upper two curves.

It will be observed that an air movement of 350 ft. per min. increases the output from 70 per cent at 90 deg. dry bulb to 55 per cent at 110 deg., when the work is based on equal increase in the pulse rate, and from 26 per cent at 90 deg. to 20 per cent at 110 deg. when based upon equal rise in rectal temperature. In so far as pulse rate is a better index of bodily reactions the first estimate represents more nearly the actual benefit derived from the movement of the air. A conservative estimate may be arrived by taking the average of the two values—namely, an increase in the output of from about 50 per cent at 90 deg. dry-bulb temperature to 40 per cent at 110 deg. Above 110 deg. dry bulb the effect of air movement is rather small because the effective temperature approaches the temperature of the body.

Importance of Air Movement

In concluding, the writers wish to emphasize the importance of air movement and its financial aspects in mines and industrial establishments, where the working efficiency is low on account of the poor working conditions. Further work is contemplated to cover the lower temperature field which will be the object of another paper in the near future.

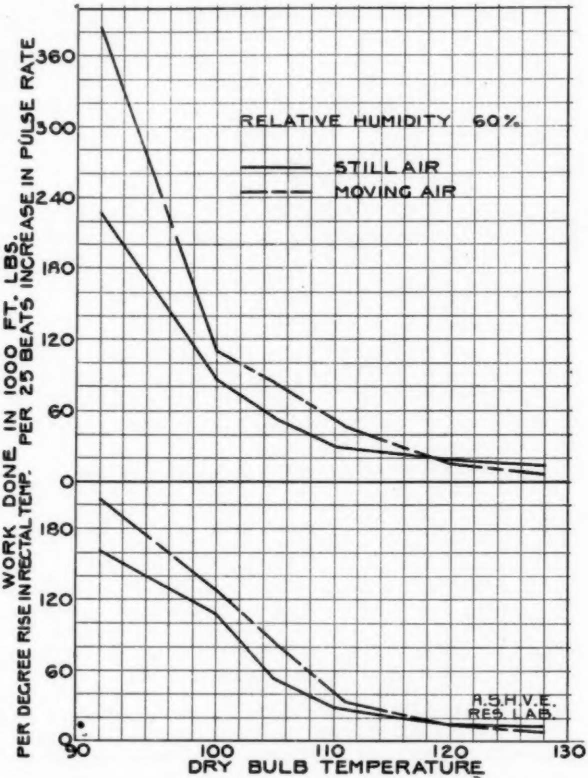


FIG. 12. AVERAGE WORK DONE BY THE SUBJECTS PER DEGREE RISE IN RECTAL TEMPERATURE, AND PER 25 BEATS INCREASE IN PULSE RATE

Acknowledgments

The authors are indebted to W. Edw. Miller, research assistant, A.S.H.&V.E. Research Laboratory and W. B. Fulton, Laboratory assistant, U. S. Bureau of Mines for their conscientious assistance in this work. In addition Mr. Fulton carried out all chloride titrations and Mr. Miller assisted in the preparations of many of the charts and tables.

No. 719

BASAL METABOLISM BEFORE AND AFTER EXPOSURE TO HIGH TEMPERATURES AND VARIOUS HUMIDITIES[1]

By W. J. McConnell[2], (Non-Member) C. P. Yaglou[3], (Member) and W. B. Fulton[4] (Non-Member)

PITTSBURGH, PA.

THE increase in body metabolism of inhabitants living in cold climates above that for warmer climates is frequently referred to in the literature, but according to Hill and Campbell[5] the evidence for this increase has, as far as they know, been empirical, resting upon increase of appetite of an individual on journeying to an Alpine or an Arctic region.

Contrary to what might be expected, metabolism also increases with exposure to high temperatures. Recently, this fact received some recognition through evidence largely drawn from experiments made on warm-blooded animals. While the little work that was done on human beings substantiates this belief, the relation between the metabolic rate and external temperature conditions remains yet to be found. Probably the main difficulty in establishing this relation lies in the fact that in high temperatures, the wet-bulb temperature of the air becomes a much more predominant factor than the dry-bulb temperature, in influencing the human body. The difficulty in evaluating the relative importance to be attached to these measurements has only recently been obviated. Air movement should also be considered, as there can be no adequate ventilation and constancy in temperature conditions without air motion.

With the development of the effective temperature scale it becomes an easy matter to study the effect of heat upon body metabolism. Effective temperature may be regarded as a relative index of the intensity of heat felt by the human body as a result of temperature of the environment, humidity, and air movement. In other words, it takes care of all three physical factors of the air, and therefore, reduces the relation to its simplest form involving only one independent variable.

---

[1] Work carried out in cooperation with the U. S. Public Health Service and the U. S. Bureau of Mines, at its Pittsburgh Experiment Station.

[2] Past-Asst. Surgeon (R), U. S. Public Health Service, and Surgeon, U. S. Bureau of Mines.

[3] Research Engineer, A.S.H.V.E. Research Laboratory.

[4] Laboratory Assistant, U. S. Bureau of Mines.

[5] Hill, Leonard, and Campbell, J. A., Observations on Metabolism during Rest and Work with Special Reference to Atmospheric Cooling Power, Medical Research Council, S. R. Series No. 73, Part 6, pp. 145–186.

Presented at the Annual Meeting of the American Society of Heating and Ventilating Engineers, Boston, January, 1925.

As part of the general program of the investigation of the physiological effects of high temperatures with various humidities undertaken by the U. S. Public Health Service and the U. S. Bureau of Mines, cooperating with the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in its laboratory at the Pittsburgh Experiment Station of the Bureau of Mines, a series of experiments were conducted for the purpose of establishing a direct correlation between the rate of metabolism and external temperature conditions. An attempt was also made to correlate the metabolic rate with the principal physiological body reactions, such as body temperature and pulse rate, in an effort to find a body index for the rate of metabolism.

While a large number of observations of the rate of oxygen consumption were made, many of these, on account of complicating influences, such as eating within a few hours before the test and unusual muscular exertion on the part of the subject, were discarded. Although samples for analyses were collected from 10 subjects, the large majority were collected from two subjects who became trained in the method used. Table 1 gives the average normal measurements for the subjects employed in the experiments, from which the body surface was determined by means of DuBois standard chart. The clothing worn by the subjects during the experiments consisted of light-weight union underwear, work trousers, and shirt, socks and shoes. These were practically the same as worn during the experiments on effective temperature.

TABLE 1. STATISTICS OF SUBJECTS OF EXPERIMENTS

Subjects	Weight in Kilos	Height in Cm.	Body Surface, Sq. Meters
W. J.	61.6	163.0	1.66
C. A. H.	64.5	183.1	1.84
R. L.	69.5	191.0	1.97
F. C. H.	72.0	171.3	1.84
C. P. Y.	59.2	167.0	1.68
W. E. M.	59.4	172.8	1.72
B. T.	71.1	174.7	1.82

Procedure

The first 100 samples of expired air were collected from subjects without any preliminary precautions. These samples were taken while the subjects were in a sitting position. Under these conditions, although the CO_2 produced and O_2 consumed invariably increased after exposure to high temperatures[6], it became quite impossible to disentangle the various factors which entered into the experiments. The remainder of the observations were made under the greatest degree of simplification, attained as follows: Each subject refrained from eating breakfast on the morning of the test. On entering the primary room they assumed a recumbent posture on stretchers, and maintained a condition of rest as absolute as possible for a period of two hours before the first sample was taken. Frequently the subjects slept during that period. After the sample was taken, each subject was carried on the stretcher into the chamber where he continued to rest in the same position. In the chamber he was exposed to a constant high temperature and humidity over a period of time varying with his ability to endure the condition. Near the end of this period another sample (and sometimes two or three) was taken before the subject left the chamber. All experiments were conducted in practically still air, the air being renewed at an adequate rate.

[6] McConnell, W. J., and Houghten, F. C., Some Physiological Reactions to High Temperatures and Humidities. JOURNAL A.S.H.V.E., March, 1923, pp. 141–144.

Method of Collecting Samples

The apparatus used in this work was constructed by the Bureau of Mines, and consists of a graduated gasometer situated outside the test chamber, connected with a mouthpiece by means of $1^1/_4$ in. rubber tubing. A quick-acting valve controls the inlet of the gasometer. The bell of the gasometer is maintained in equilibrium with the incoming air. Considerable difficulty was experienced in obtaining a mouthpiece with valves suited for the tests. Several types of valves are available which, though satisfactory for use in certain breathing apparatus, could not be used in these experiments because of either a small amount of air leakage, or because of the impractical position in which the valve had to be held in order to function. After some experimentation a valve[7] was developed and successfully used in the collection of the samples.

The subject breathed through the gasometer for several minutes before the sample was taken in order to become accustomed to the apparatus, while at the same time

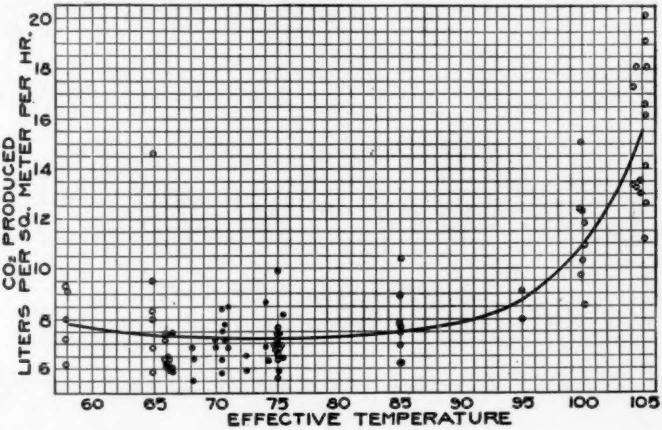


FIG. 1. RELATION BETWEEN EFFECTIVE TEMPERATURE AND LITERS OF CO_2 EXPIRED BY EACH SUBJECT PER SQ. METER OF BODY SURFACE PER HOUR

the exhalations forced out the stagnant air in the system. Approximately 60 liters of expired air was collected from the subject in each sample. This was determined by multiplying the factor (which was found by calibration to be 0.0992 liters per centimeter rise of the bell) of the gasometer by the number of centimeter rise of the bell. All volumes were reduced to 0 deg. cent. temperature and dry, and 760 mm. barometric pressure. The ventilation rate or volume per minute was obtained by dividing the total volume by the time in minutes. From each sample collected an average sample was analyzed on a large Haldane apparatus[8] for CO_2 and O_2, from which the heat developed within the body, under the various external conditions, was computed using Zung's table of calorific equivalents of 1 liter of oxygen.

[7] An Improved Air Valve for Apparatus Used in Basal Metabolic Work, by W. B. Fulton, *Archives of Internal Medicine*, vol. 33, April, 1924, pp. 497–499.
[8] The Sampling and Examination of Mine Gases and Natural Gas, by G. A. Burrell and F. M. Seibert, *Bull.* 42, Bureau of Mines, 116 pp. Revised in 1924 by G. W. Jones (in press).

Data and Results

While it is not the purpose of this paper to discuss the general principles of metabolism or to review the enormous amount of literature on the subject which has been collected from many sources, the reader's attention is invited to other recent investigations along these lines. Moss,[9] who is making a study of the subject, found an increase in food-consumption with temperature and contemplates further experimentation on the exact cause of the increased metabolism. Barcroft and Marshall[10] have also carried on some recent experiments to determine the effect of exposure to heat. Under the conditions of these latter experiments, no commensurate rise in the O_2 consumed was found. Unfortunately, the severity of the exposure was recorded in dry-bulb temperature only; but judging from the

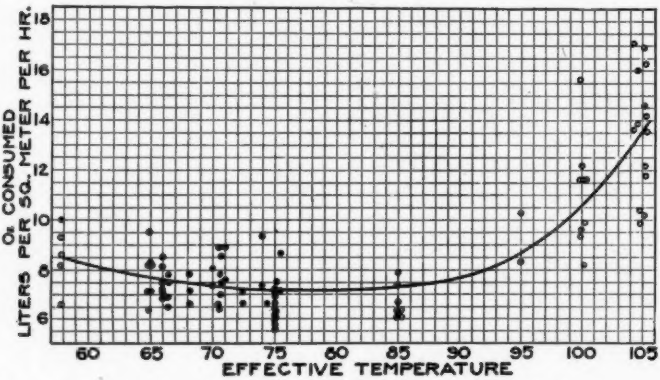


FIG. 2. RELATION BETWEEN EFFECTIVE TEMPERATURE AND LITERS OF O_2 CONSUMED BY EACH SUBJECT PER SQ. METER OF BODY SURFACE PER HOUR

pulse rates obtained, the wet-bulb readings were low and therefore the effective temperature was below that where a noticeable increase in metabolism would be expected.

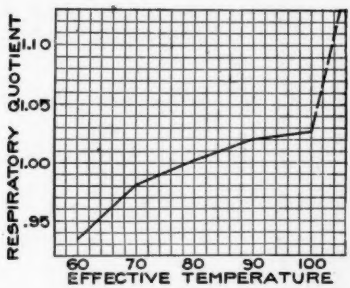
The data and results of the present series of experiments are presented in Table 2. With the exception of few experiments, initial expiratory samples were taken in the primary room, the temperature conditions of which are given in the left of Column 3. The test chamber, or secondary room conditions are given in the right of the same column, and the time of exposure before taking samples is shown in Column 4. Columns 5 and 6 give respectively the CO_2 produced and the O_2 consumed in liters per hour at 0 deg. cent. temperature and 760 mm. of mercury barometric pressure. The ratio of the former to the latter, or respiratory quotient, is computed in Column 7. The total number of calories, developed per hour is given in Column 8, from which the heat produced within the body per square meter of body surface per hour is calculated in Column 9. Columns 10 and 11 give the average physiological reactions recorded during the period of time in which the samples were taken.

[9] Moss, Prof. K. Neville, Some Effects of High Air Temperature upon the Miner. Sixth Report to Committee on the Control of Atmospheric Conditions in Hot and Deep Mines, Annual General Meeting, *Institution of Mining Engineers* (England), Nov. 29, 1923.

[10] Barcroft, Jos., and Marshall, E. K., Fr., Note on the Effect of External Temperature on the Circulation in Man. *Jour. of Physiology*, vol. LVIII, No. 2 and 3 Dec 28, 1923, pp. 145–156.

An examination of this table will disclose that both the CO_2 produced and the O_2 consumed increase with exposure to temperatures either higher or lower than the normal atmospheric condition. The range of temperature employed in the experiments varied from about 55 to 130 deg. effective temperature, but the table includes results up to about 105 deg. effective temperature only. To obtain a fair sample it was found necessary that the subjects should be exposed to the constant temperature conditions for a period of at least an hour before taking respiration samples. For temperature higher than 105 deg. effective temperature, the subjects of the experiments could not endure the condition an hour, and the results obtained at these higher temperatures were consequently too low. Apparently the human mechanism did not have enough time to react fully to the temperature environment, and for this reason a number of samples that were taken after a short period of exposure were discarded. Figs. 1 and 2 show respectively the variation of CO_2 produced and O_2 consumed, with effective temperature. To afford a uniform basis of comparison between different subjects, these two quantities are expressed in liters per square meter of body surface per hour. Black circles represent observations made in the primary room, while the white or open circles those made in

FIG. 3. RELATION BETWEEN RESPIRATORY QUOTIENT AND EFFECTIVE TEMPERATURE



the secondary or test room. It will be observed that the curves are very similar and have the same characteristics. They both attain a minimum value of 7.2 liters per square meter of body surface per hour, within a temperature zone between 70 and 85 deg. effective temperature, where the rate of gaseous exchange is practically constant. Above and below this zone, however, both quantities increase at an accelerated rate. At the comfortable temperature of 65° effective temperature the figures show an average of 7.3 liters of CO_2 expired and 7.7 liters of O_2 consumed per square meter of body surface per hour. This corresponds to a respiratory quotient of 0.948.

It is of interest to note that the rate of gaseous exchange increases rapidly above 85 deg. effective temperature, and with a still higher rate after the body temperature is passed.

The respiratory quotient or the ratio of $\frac{CO_2}{O_2}$, in these experiments varied from about 0.84 to 1.55. Fig. 3 shows the relation of this ratio to effective temperature, computed from points on the curves as drawn in Figs. 1 and 2. As the temperature increases, the respiratory quotient increases approximately at the same rate, until at about 80 deg. effective temperature it becomes unity. In other words, the CO_2 produced becomes equal to the O_2 consumed in respiration. From 80 deg. to about body temperature the variation in the respiratory quotient is rather small,

TABLE 2. DATA AND RESULTS

Test No. and date	Subjects	Test Room Conditions: Primary: Dry bulb	Primary: Wet bulb	Primary: Effective temp.	Secondary: Dry bulb	Secondary: Wet bulb	Secondary: Effective temp.	Exposure Before Taking Sample: Primary room: Hours	Primary room: Minutes	Secondary room: Hours	Secondary room: Minutes	CO_2 produced liters per hour	O_2 consumed liters per hour	Respiratory quotient	Tota calories per hour	Calories per sq. meter of body surface per hour	Rectal temperature, °F.	Pulse rate, beats per min.
37 A. S. 6-4-23	W. J.	75.0	66.8	70.5				2	00			9.60	10.92	0.879	53.5	32.6	98.0	72
	W. J.				105.6	104.2	104.3			2	00	22.14	22.50	0.984	113.1	69.2	102.1	158
	C. A. H.	75.0	66.8	70.5				2	20			15.36	16.44	0.934	81.6	44.3	98.4	64
	C. A. H.				105.6	104.2	104.3			0	56	31.74	31.44	1.009	158.9	87.7	102.5	158
38 A. S. 6-8-23	W. J.	76.4	64.2	70.0				2	00			11.76	13.38	0.878	65.6	39.5	98.0	68
	W. J.				100.0	100.0	100.0			1	37	16.86	15.84	1.063	81.1	49.4	101.7	132
	C. A. H.	76.4	64.2	70.0				2	25			12.48	13.44	0.929	66.7	36.8	98.2	60
	C. A. H.				99.9	99.9	99.9			1	14	22.62	22.50	1.005	113.7	62.7	101.8	150
39 A. S. 6-11-23	W. J.	77.0	69.0	72.5				2	00			9.66	11.10	0.870	54.3	33.2	98.0	80
	W. J.				95.0	95.0	95.0			1	54	13.20	13.74	0.961	68.6	41.9	99.3	96
	C. A. H.	77.0	69.0	72.5				2	20			11.88	13.08	0.908	64.6	35.6	99.0	76
	C. A. H.				95.0	95.0	95.0			2	1	16.62	18.90	0.879	92.6	51.2	100.1	96
40 A. S. 6-27-23	W. J.	76.0	67.0	71.0				1	30			11.22	12.60	0.890	61.9	37.4	98.4	72
	W. J.				104.6	104.6	104.6			1	2	29.82	26.34	1.130	137.5	83.8	102.0	140
	C. A. H.	76.0	67.0	71.0				1	50			15.66	16.32	0.960	81.5	43.3	98.0	70
	C. A. H.				104.6	104.6	104.6			0	54	24.48	25.44	0.963	127.8	69.5	101.8	145
41 A. S. 6-29-23	W. J.	79.5	63.0	70.6				1	30			10.32	10.56	0.978	52.9	32.1	98.0	74
	W. J.				105.3	105.3	105.3			1	00	23.16	22.26	1.040	113.5	68.7	100.8	114
	W. J.				105.3	105.3	105.3			1	20	27.42	25.20	1.087	129.3	79.3	101.9	136
	C. A. H.	79.5	63.0	70.6				1	55			13.74	14.28	0.962	71.4	39.4	99.0	70
	C. A. H.				105.3	105.3	105.3			0	54	23.22	22.50	1.032	114.4	63.2	100.6	128
42 A. S. 7-2-23	W. J.	77.2	65.3	70.7				1	30			11.94	12.36	0.966	61.9	37.7	97.8	80
	W. J.				105.1	105.1	105.1			0	58	18.48	16.80	1.100	86.8	52.9	100.8	120
	W. J.				105.1	105.1	105.1			1	18	33.36	27.84	1.200	147.1	88.6	102.3	136
	C. A. H.	77.2	65.3	70.7				1	50			14.40	15.90	0.906	78.4	43.3	98.0	82
	C. A. H.				105.1	105.1	105.1			0	53	29.88	26.88	1.112	130.3	75.6	101.6	144
43 A. S. 7-6-23	R. L.	79.7	72.8	75.5				1	30			12.48	14.04	0.890	69.0	35.2	98.0	64
	R. L.				140.0	93.2	100.1			1	21	16.68	16.02	1.041	81.6	41.7	101.1	105
	R. L.				140.0	93.2	100.1			2	10	21.36	19.20	1.112	99.4	50.7	102.3	112
	C. A. H.	79.7	72.8	75.5				1	50			15.12	15.90	0.951	79.3	43.2	98.0	72
	C. A. H.				140.0	93.2	100.1			1	21	21.78	21.48	1.013	108.6	59.1	101.1	114
44 A. S. 7-1023	F. C. H.	80.7	69.8	74.3				2	00			11.58	12.30	0.942	61.2	33.3	99.0	70
	F. C. H.				104.8	104.8	104.8			1	00	23.70	19.02	1.246	101.6	55.2	101.3	172
	F. C. H.				104.8	104.8	104.8			1	15	24.78	17.94	1.381	98.7	53.7	102.2	186
46 A. S. 7-13-23	C. A. H.	81.8	70.8	75.1				2	00			13.56	13.68	0.992	68.9	38.1	98.6	66
	C. A. H.				105.3	105.3	105.3			0	50	33.18	26.10	1.269	140.3	76.2	101.7	134
	C. A. H.				105.3	105.3	105.3			1	10	35.22	21.84	1.612	126.3	69.8	102.4	140

49 A. S. 7–20–23	W. J.	82.1	68.2	74.0				1	35			11.22	12.12	0.925	60.1	36.5	98.2	75
	W. J.				100.0	99.9	99.9			0	56	16.02	15.30	1.047	78.0	47.6	100.3	104
	C. A. H.	82.1	68.2	74.0				1	55			15.90	17.28	0.920	85.5	47.3	98.4	60
	C. A. H.				100.0	99.9	99.9			0	40	22.74	21.24	1.071	108.0	59.7	99.7	110
	C. A. H.				100.0	99.9	99.9			1	35	27.66	28.80	0.960	143.9	79.5	102.1	160
50 A. S. 7–23–23	W. J.				65.3	64.2	64.9			1	10	9.60	10.32	0.930	51.2	31.3	97.9	58
	W. J.				65.3	64.2	64.9			3	16	13.14	13.44	0.977	67.4	41.2	96.8	56
	C. P. Y.				65.3	64.2	64.9			0	47	13.92	11.82	1.178	61.1	36.5	99.2	68
	C. P. Y.				65.3	64.2	64.9			3	46	24.48	15.84	1.549	90.3	53.4	97.6	62
	W. E. M.				65.3	64.2	64.9			0	18	11.52	12.00	0.960	60.0	35.3	98.0	70
	W. E. M.				65.3	64.2	64.9			3	14	16.08	13.98	1.150	73.1	43.3	97.3	68
51 A. S. 7–24–23	W. J.				75.2	74.7	75.0			1	31	9.66	9.30	1.039	47.4	29.2	98.2	64
	W. J.				75.2	74.7	75.0			4	36	10.38	10.08	1.029	51.2	31.2	98.0	60
	R. L.				75.2	74.7	75.0			1	55	10.98	11.16	0.985	56.1	28.7	97.9	64
	R. L.				75.2	74.7	75.0			5	00	13.50	11.70	1.152	61.3	31.3	97.6	64
	W. E. M.				75.2	74.7	75.0			2	4	11.70	11.76	0.995	59.3	35.0	98.2	63
	W. E. M.				75.2	74.7	75.0			5	11	11.76	12.12	0.970	60.7	35.8	97.8	68
52 A. S. 7–25–23	W. E. M.				84.7	68.3	75.0			2	14	11.22	10.62	1.057	54.4	32.0	98.1	70
	W. E. M.				84.7	68.3	75.0			3	59	16.86	11.82	1.427	65.7	38.9	97.8	64
	W. E. M.				84.7	68.3	75.0			4	57	13.08	11.34	1.153	59.4	35.1	97.8	64
	R. L.				84.7	68.3	75.0			1	58	13.38	12.00	1.115	62.2	31.7	98.0	62
	R. L.				84.7	68.3	75.0			3	45	14.46	14.04	1.030	71.4	36.5	98.4	64
	R. L.				84.7	68.3	75.0			4	43	13.98	12.30	1.138	64.1	32.9	98.4	66
53 A. S. 7–27–23	R. L.				102.3	77.8	85.0			2	32	12.24	12.06	1.015	61.0	31.1	98.6	70
	R. L.				102.3	77.8	85.0			5	8	12.24	11.70	1.048	59.7	30.5	98.6	70
	W. E. M.				102.3	77.8	85.0			2	2	12.72	11.28	1.129	58.6	34.5	98.7	68
	W. E. M.				102.3	77.8	85.0			4	52	15.12	13.38	1.130	68.0	39.7	98.5	70
54 A. S. 7–28–23	R. L.				85.0	85.0	85.0			1	23	13.56	11.88	1.141	62.0	31.7	98.1	66
	R. L.				85.0	85.0	85.0			4	11	15.06	12.24	1.230	65.2	33.3	97.8	64
	W. E. M.				85.0	85.0	85.0			1	22	17.76	12.60	1.410	69.8	40.8	98.5	72
	W. E. M.				85.0	85.0	85.0			4	6	13.20	10.62	1.243	56.7	33.6	98.2	66
1 E. 7–10–24	W. J.	74.5	62.5	68.2				1	30			10.56	11.76	0.897	57.9	35.3	98.2	72
	W. J.				67.6	64.3	66.0			0	59	10.02	11.22	0.893	55.2	33.6	97.7	66
	W. J.				67.6	64.3	66.0			2	21	10.50	11.64	0.902	57.4	35.0	97.5	60
	B. T.	74.5	62.5	68.2				1	40			10.14	12.12	0.837	58.8	32.0	98.9	62
	B. T.				67.6	64.3	66.0			1	26	11.28	13.26	0.851	64.5	35.1	98.2	58
	B. T.				67.6	64.3	66.0			2	48	11.76	13.86	0.848	67.3	36.6	98.0	57
	W. E. M.	74.5	62.5	68.2				1	50			11.58	13.44	0.862	65.6	38.7	98.4	78
	W. E. M.				67.6	64.3	66.0			1	49	12.24	13.86	0.883	68.0	40.3	98.4	70
	W. E. M.				67.6	64.3	66.0			3	7	12.54	14.46	0.867	70.7	41.8	98.2	69
2 E. 7–11–24	W. J.	73.7	59.2	66.5				1	10			9.72	10.74	0.905	53.0	32.3	98.0	72
	W. J.				59.4	55.7	57.9			1	3	11.88	13.50	0.880	66.2	40.3	98.0	56
	B. T.	73.7	59.2	66.5				1	40			10.98	12.66	0.867	61.8	33.6	98.4	59
	B. T.				59.4	55.7	57.9			1	31	11.34	12.12	0.935	60.2	32.7	98.3	52
	B. T.				59.4	55.7	57.9			2	57	17.28	17.10	1.012	86.7	47.2	98.4	54
	W. E. M.	73.7	59.2	66.5				1	55			12.72	13.32	0.955	66.5	38.8	98.4	76
	W. E. M.				59.4	55.7	57.9			2	10	13.50	14.58	0.926	72.3	42.7	97.8	68
	W. E. M.				59.4	55.7	57.9			2	52	15.48	17.04	0.908	84.1	49.6	97.7	68

but when the latter temperature is exceeded a very sudden increase is apparent, according to the limited results available at these high temperatures. Because of the insufficient number of determinations this portion of the curve is shown dotted.

Ordinarily, the respiratory quotient recorded by various investigators, within the normal range of temperature conditions, seldom exceeds unity. The question now arises how high temperatures affect the human body so as to raise the respiratory quotient above unity. Theoretical considerations suggest that free oxygen is available in the body through its liberation during the transformation of carbohydrates into fats. A study of the respiratory exchanges of animals which are rapidly laying on a store of fat at the expense of a carbohydrate diet indicates that oxygen is set free. Thus the marmon, towards the end of the summer, eats large quantities of carbohydrate food, and very rapidly lays on a thick layer of subcutaneous fat to last it during the winter.

Starling[11] points out that fat formed from carbohydrate incurs a considerable loss of oxygen. For example: he states that if glucose were entirely oxidized in the body, the amount of O_2 absorbed would be exactly equal to the amount of CO_2 involved. Thus:

$$C_2H_{12}O_6 + 6O_2 \longrightarrow 6CO_2 + 6H_2O$$

In this case the respiratory quotient would be

$$\frac{6CO_2}{6O_2} = 1$$

If, however, O_2 is being set free by the conversion of part of the carbohydrate into fat, this O_2 will be available for the oxidation of other portions of the carbohydrates. The animal will not require so much O_2 from external sources for the production of the same amount of CO_2, and therefore the CO_2 output of the animal will be greater than its O_2 intake. Pembrey[12] has shown that under this condition the respiratory quotient may be as high as 1.50.

Fig. 4 shows the calories of heat produced within the body per square meter of body surface, plotted against effective temperature. The heat produced was not measured, but was calculated from the O_2 consumption. The variation of this quantity with effective temperature is of course similar to that observed in Figs. 1 and 2. At the normal temperature of 65 deg. the average subject of the experiments developed 38.2 calories per square meter of body surface per hour. This value checks very closely with DuBois standard for basal metabolism—namely, 38.6 calories—but the curve shows that it is by no means the minimum metabolism.

It will be observed that there is a temperature zone of minimum metabolism, between 75 and 83 deg. effective temperature, within which the lowest value of 36 calories per sq. meter per hour is reached. It is the writers' belief that *basal metabolism should be measured within this zone.* The belief is substantiated from results of various other investigators who recorded values well below DuBois standard, depending upon the temperature in which the observations were made.

Apparently little importance has been attributed to the surrounding temperature conditions of the subjects in previous experiments, and this is one of the points the writers propose to emphasize through the evidence presented herein.

---

[11] Principles of Human Physiology, by Ernest H. Starling, 3rd edition, 1920, Section III, pp. 826-838.

[12] Work cited by Starling, p. 830, *Ibid.*

Attention is called to the large temperature range of successful operation of the body thermostatic control, which adjusts the heat production to accurately balance the heat loss, according to the temperature of the environment. This is represented by the flat portion of the curve for ordinary atmospheric conditions. Above 85 deg. effective temperature, however, there is apparently a strain on the mechanism. The body makes strenuous efforts to resist rise in its temperature by promoting evaporation of perspiration from its surface, but the limit of the compensating action of the thermostatic control is reached, and the latter fails entirely above 90 deg. effective temperature. This is indicated by the rapid increase of heat production at the higher temperatures. At 105 deg. effective temperature the heat production is twice as great as at the normal temperature of 65 deg.

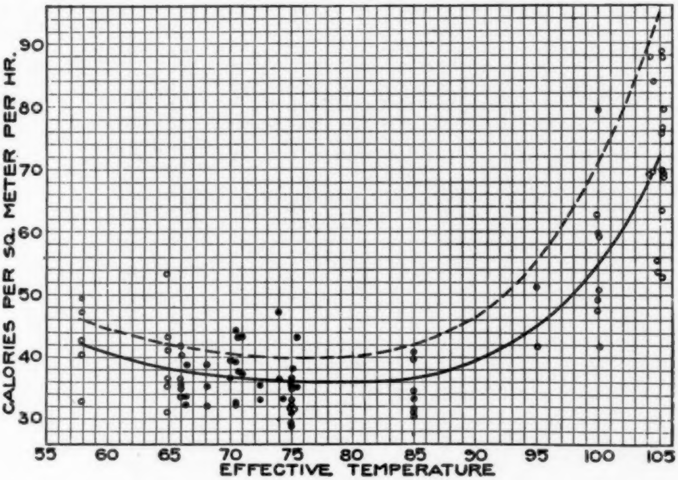


Fig. 4. Calories Produced per Sq. Meter of Body Surface per Hour at Various Effective Temperatures

A tendency for an increase in heat production is also shown below 56 deg. effective temperature, which heat is necessary to keep the body warm in cold weather.

Comparison of Test Results

It is of interest to introduce here an analysis made by one of the writers of the previous results obtained by O_2 consumption very recently in the psychrometric chambers of the Research Laboratory of the American Society of Heating and Ventilating Engineers.[13] The upper dotted curve in Fig. 4 represents the normal metabolism at different temperatures, humidities and air velocities. The last three factors are represented here by the effective-temperature index. As the experiments under consideration were conducted practically under the same conditions as those presented in this paper, the differ-

[13] The Heat Given up by the Human Body and its Effect on Heating and Ventilating Problems. by C. P. Yaglou, Journal A.S.H.&V.E., Aug., 1924, pp. 597–609.

ence between the two curves represents the increase in metabolism due to food and sitting position. The two curves are practically parallel within the ordinary range of temperature and their difference amounts to about 4 calories per sq. meter per hour. In other words, the metabolism at the normal temperature of 65 deg. effective temperature increased by 11 per cent over the basal value when the subjects partook of their regular diet and were sitting comfortably on chairs.

For higher temperatures exceeding 80 deg. effective temperature the increase in metabolism due to food and sitting position is not constant, but is accelerated, as shown by the divergence of the two curves. The reason for this may be due to the fact that in one case the basal metabolism is at the expense of substances

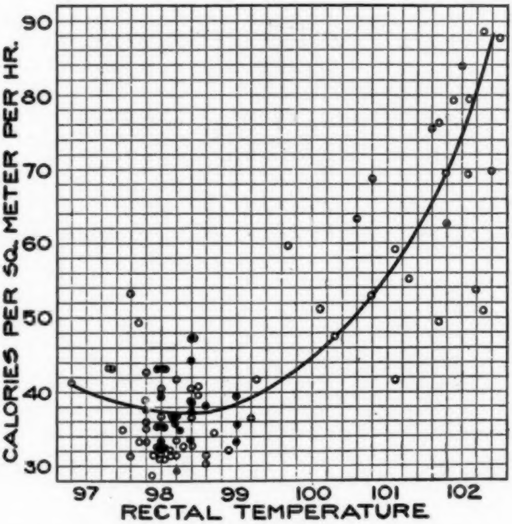


FIG. 5. CORRELATION BETWEEN HEAT PRODUCTION AND TEMPERATURE RECTAL

stored within the body, while in the other case there is material in the form of newly ingested food available for chemical transformation.

An examination of Table 2 will show that wherever more than one sample was taken in the test chamber the heat production invariably increased with the time of exposure. This is to be expected when considering that the physiological reactions vary with temperature and time of exposure. Accordingly, an attempt was made to correlate the rate of metabolism to rectal temperature and pulse rate in Figs. 5 and 6 respectively.

Fig. 5 shows that heat production is minimum at a body temperature of about 98.4 deg. fahr. and that it increases both above and below this temperature. It stands to reason that metabolism should increase when there is a drop in body temperature, to keep the body warm. The increase for the higher temperatures is attributed to the warming up of the cells, and the marked rapid increase for

temperatures above 100 deg. is apparently due to the breaking down of the human thermostatic control.

Similarly in Fig. 6 it is found that the heat production attains a minimum value of about the same magnitude as in the previous figure, namely 37.0 cal. per sq. meter per hour, at a pulse rate of 68 heats per minute. Metabolism again increases with higher or lower pulse rates but the rate of increase is not as great as it is with temperature. A comparison of the two figures shows that the pulse curve is much flatter than that for temperature, indicating that pulse rate is a more dependable direct index of the metabolic rate.

Summary

In summarizing the results of these experiments the writers wish to emphasize the significance of the dry- and wet-bulb temperature, and movement of air. These

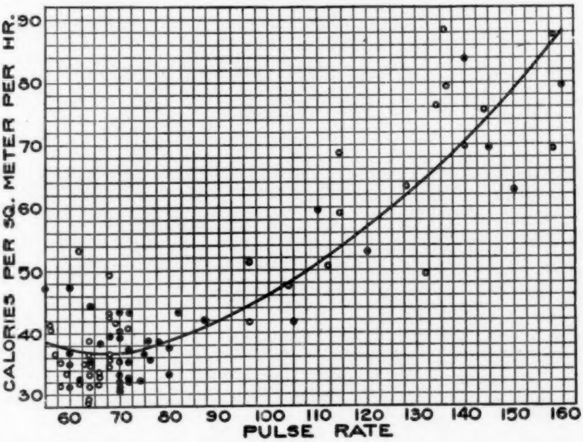


FIG. 6. CORRELATION BETWEEN HEAT PRODUCTION AND PULSE RATE

three factors have been shown to be reducible to one index, called "effective temperature," which is determined from the above 3 readings, using an effective-temperature chart or table. The problem is thus greatly simplified and the effect of various other factors can be studied independently of temperature.

Conclusions

Some of the important conclusions drawn from this study are:

1. The rate at which carbon dioxide is produced and oxygen is consumed increases with exposure to either high or to low temperature.
2. Heat production increases with exposure to high and low temperature.
3. There is a zone of minimum metabolic rate between 75 and 83 deg. effective temperatures, within which basal metabolism should be measured.
4. The metabolic rate becomes excessive when the temperature of the environment is higher than body temperature.
5. The rate of metabolism correlates fairly well with body temperature and pulse rate.

No. 720

PROGRESS REPORT ON THE CRITICAL VELOCITY OF THE FLOW OF STEAM IN TWO PIPE SYSTEMS

Investigated by Louis Ebin and Gordon Eisenhart

Reported by Margaret Ingels, Pittsburgh, Pa.

MEMBER

This investigation was started by Louis Ebin who left the Research Laboratory staff to accept a position elsewhere. The work was then carried on by Gordon Eisenhart until his sudden death unfortunately stopped research on the problem before the complete results were obtained and conclusions drawn.

The efficient and painstaking efforts of the investigators are evident by the data compiled and this report is a record of the results of that part of the investigation which had been completed and includes some outstanding conclusions which are shown by the data.

THE results given in this report are those obtained from 627 tests on flow of steam in vertical, horizontal and inclined pipes of several diameters. It partly covers that phase of the work given in the plan[1] of work on steam flow as:

Critical Velocity of Risers—two pipe systems.

Critical Velocity of Horizontal Pipes, all degrees pitch.

Figs. 1 and 2 give the set-up with which these investigations were made.

The data taken in all these tests consist of the following:

	Data Recorded	Tests Made		
		208 tests	90°	pitch 1.027" dia.
1.	Length of Test	103 tests	90°	pitch 0.84" dia.
2.	Inside Diameter of Pipe	108 tests	90°	pitch 1.36" dia.
3.	Pitch of Pipe	27 tests	90°	pitch 1.59" dia.
4.	Length of Pipe	47 tests	0°	pitch 1.027" dia.
5.	Header Pressure	30 tests	2°	pitch 1.036" dia.
6.	Riser Pressure Drop	27 tests	−2°	pitch 1.036" dia.
7.	Condensate in Return Riser	19 tests	−3.78°	pitch 1.037" dia.
8.	Condensate in Supply Riser	23 tests	10°	pitch 1.037" dia.
9.	Velocity of Steam (from R. R. condensate)	16 tests	−2.7°	pitch 1.017" dia.
10.	Velocity of Steam (from total condensate)	14 tests	3°	pitch 1.017" dia.

[1] Capacity of Steam Heating Risers as Affected by Critical Velocity of Steam and Condensate Mixtures, F. C. Houghten, Louis Ebin, Journal A.S.H.&V.E., March, 1923.

Presented at the Annual Meeting of the American Society of Heating and Ventilating Engineers, Boston, Mass., January, 1925.

Explanation of Data.

1. The length of a test varies from 9 to 42 min., and readings of all the variables are taken every 3 min.

2. The inside diameter of the pipe is measured in inches at both ends, and the average used as the inside diameter of the pipe.

3. The pitch of the pipe is given in degrees from the horizontal. That is, a horizontal pipe has a pitch of 0 deg., the vertical pipe a pitch of 90 deg.

4. The length of the pipe is given, both for the overall, and for the distance between the points where the pressure drops are read. The length of the pipe

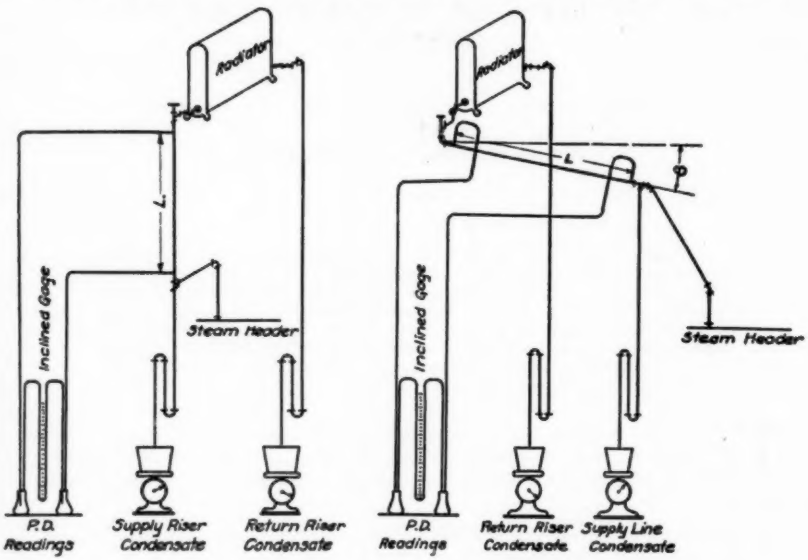


Fig. 1. Fig. 2.

CONNECTIONS FOR DETERMINING CRITICAL VELOCITY IN TWO PIPE SYSTEMS

between the points where the pressure drops are taken is corrected to 10 ft. lengths in the tables of results.

5. The header pressure is given in inches of water, and varies from 0.10 to 10.0 in. The header pressure is used as a gage for varying the flow of the steam in the risers, but has no significance in the results.

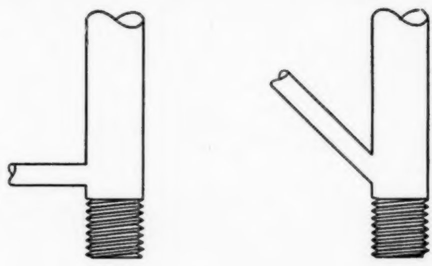
6. The pressure drop in the riser is given in inches of water, and the connections for the gage are made so that the entrance losses are presumably eliminated. A sketch of the methods of measuring the pressure difference is given in Fig. 3. It is very difficult to get the exact pressure drop in the riser, and it is very important to get it accurately. The connection shown in *a* of Fig. 3 was the style used in the early tests, but the style shown in *b* was adopted because it was thought that any tendency towards the water getting into the connections to the gage was lessened by making it possible for the water to drain back into the supply line.

7. The condensate in the return risers is that which drained into the return line bucket as shown in Fig. 1, and Fig. 2, and is measured in pounds and ounces. The condensate is corrected to the rate per hour.

8. The condensate in the supply riser is that which drained into the buckets connected to the supply risers as shown in Fig. 1, and Fig. 2, and corrected to the rate per hour. It is quite obvious that as the velocity of the steam increases, the amount of condensate returning from the supply riser decreases for two reasons. *First*, the steam does not stay in contact with the walls of the pipe for the radiation to cause as much condensation, and *Second*, because the steam at high velocities will carry the condensate along with it.

9. Velocity of the steam figured from the condensate in the return riser should be the velocity of the steam at the outlet of the riser being tested, although there

Fig. 3. Pressure Gage Connection



are numerous factors that may keep this from being the case. The condensate in the supply riser is usually very small, but it should not be neglected.

10. The velocity of the steam figured from the sum of the condensates in the supply riser and return riser should be the velocity of the steam entering the riser which is being tested.

Formula for figuring the velocity of steam:

V = velocity of steam in feet per second
C = condensate in pounds per hour
D = density of steam
d = inside diameter of pipe in inches

$$V = \frac{C}{60 \times 60 \times D\ {}^1/_4\ \frac{\pi d^2}{144}} = \frac{C}{0.7342\ d^2}$$

The use of the glass tube as a riser in the tests on the one-pipe system showed so much of what takes place within the riser, that it was again tried in the tests on the two-pipe systems. The curve in Fig. 4 shows the data obtained from the tests on the glass tube. The notations on the curve show the influence of the velocity of the steam on the visible moisture. At low velocities the condensate flows down the tube but as the velocity of the steam increases the drops of condensate are caught up and cause a disturbance at the entrance of the tube. When there is a sufficiently high velocity of the steam to carry the condensate with it there is no further disturbance at the entrance. The region where the velocity is catching enough drops to cause a turbulence to where the velocity entirely changes the direction of the

condensate is known as the region of critical velocity. The results of these tests cannot be compared with values obtained from Babcock's, Unwin's or Carpenter's formulae for the velocity of steam in pipes, because their formulae are presumably for dry steam.

The curves, Figs. 5 to 11, inclusive, showing the results of the tests, may be divided into three classes: *One*, those in which the flow of the condensate is first counter flow, changing to co-flow with the steam in the pipes, as the steam velocity

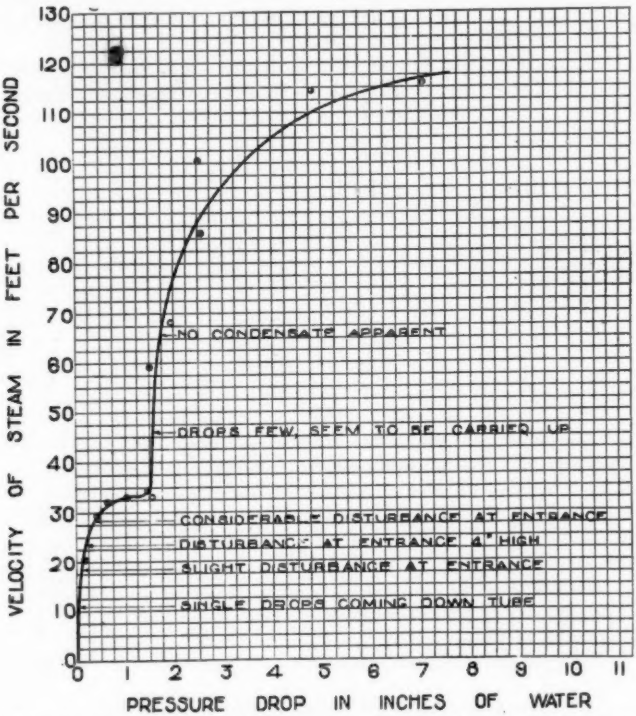


FIG. 4. CURVE SHOWING RELATION BETWEEN VELOCITY OF STEAM AND PRESSURE DROP IN GLASS TUBE

increases; that is, the results of the tests in which the steam was flowing against the condensate; *Two*, those in which the condensate is carried along with the steam by virtue of the steam velocity entirely; that is, horizontal pipes; *Three*, those in which the condensate flows with the steam by virtue of its own weight in addition to being carried by the velocity of the steam; that is, the results of the tests in which the steam is flowing with the condensate.

Discussion

The curves will be discussed in groups, according to the above classification.

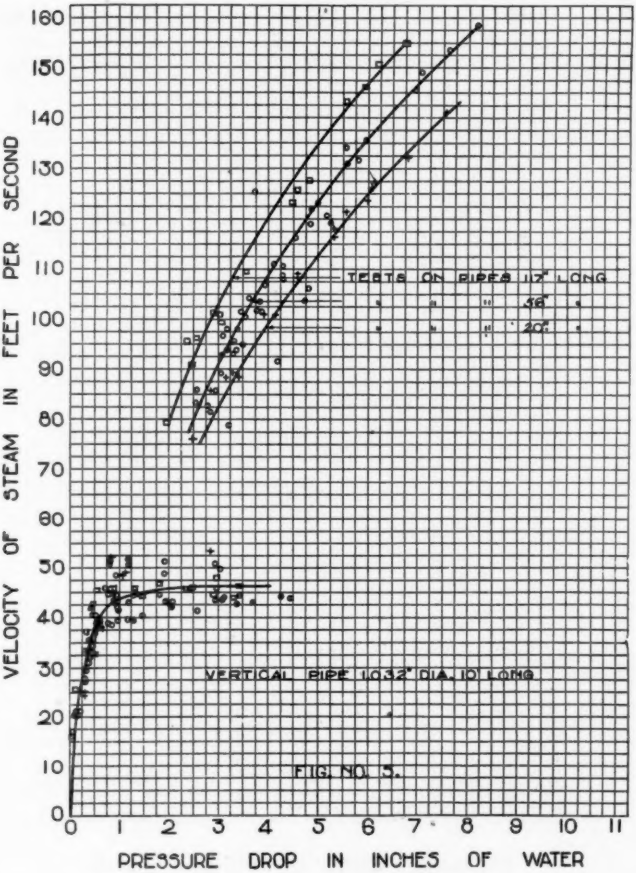


FIG. 5. RESULTS OF TESTS (CLASS 1)

Class One—Curves in Figs. 5, 6, 7 and 9.

The characteristics obtained by the tests on the glass tube are shown in all these tests. Peculiar disturbances occur in the risers between the velocities of steam at 30 ft. per second and steam at 80 ft. per second, for the three sizes of pipes considered. The smaller the pipe, the lower is the velocity for these disturbances. This is possibly due to the condensate having a greater *depth* on the perimeter of the smaller pipe, making the free area of the pipe less, and causing a greater actual velocity of steam than is figured from using the full area of the pipe for steam flow.

The shorter the pipe tested, and corrected to a 10 ft. length, the greater is the pressure difference for the same velocity. If this is true, tests would have to be

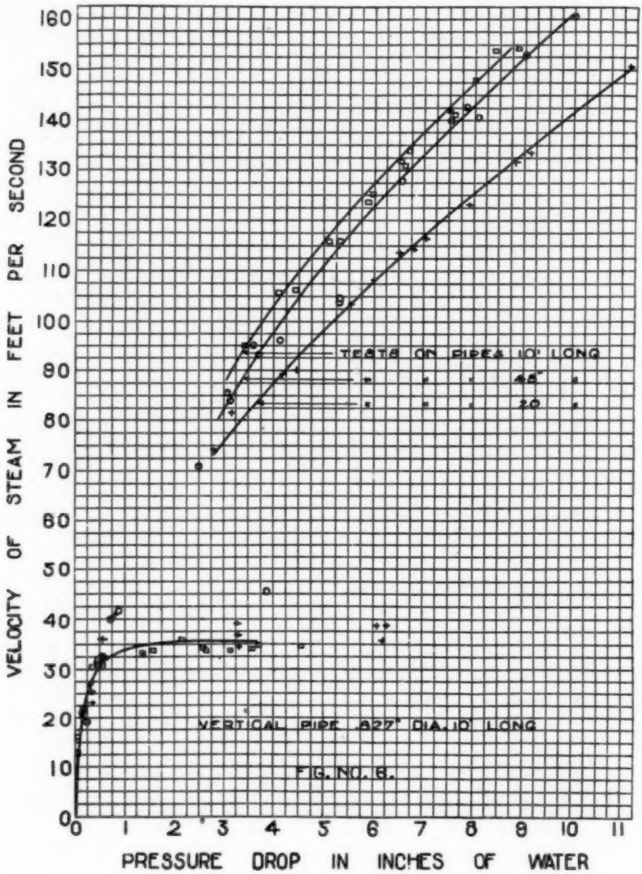


FIG. 6. RESULTS OF TESTS (CLASS 1)

made on pipes of every length, and for each diameter to determine the relation of pressure drop to velocity. There is a probability that the variation is due to the disturbances at the lower end of the riser. The magnitude of this variation would depend on how far the connection for the gage is above the end of the pipe, on how far the turbulence is above the entrance of the pipe, on how wet the steam is that enters the pipe, and how much condensate is returning through the supply riser. If none of the end turbulence of the riser affects the gage there should be a direct relation between the pressure drop and the length of the pipe for a given velocity. The consistent variation according to the lengths of the pipes tested indicates that there is some end loss measured in the pressure drop.

Consider the curves shown in Fig. 9. The region of critical velocity begins at a lower velocity for the pipes with the lower degree of slope; that is, the velocity of steam necessary to change the direction of the condensate varies with the slope of the pipe. This can be explained by the falling force of the water being less in the pipes with the smaller slopes, therefore less steam pressure is required to overcome this force and change the direction of flow of the condensate. The upper portions

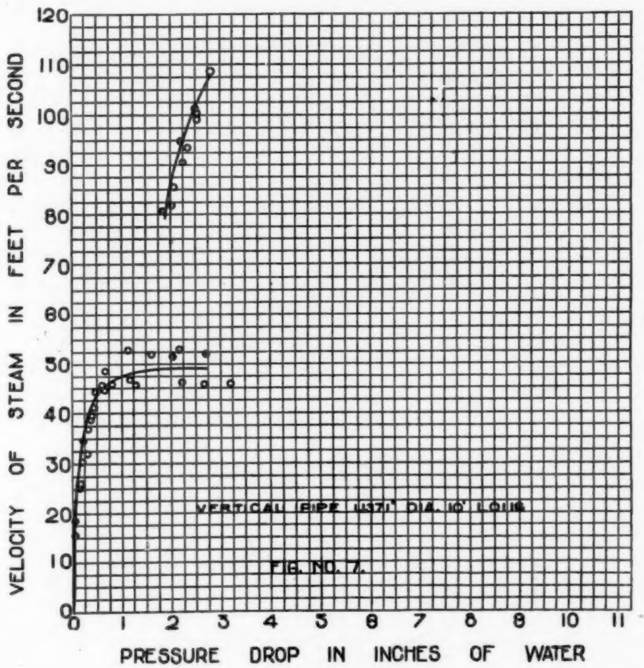


FIG. 7. RESULTS OF TESTS (CLASS 1)

of these curves would probably coincide if there had been sufficient test made to obtain a truer average of the points.

Class Two—Curves in Fig. 8.

In this figure are shown two curves for the flow of steam in horizontal pipes. The shorter lengths of pipes corrected to a 10 ft. length show greater pressure drops for the same velocity than longer pipes, and again this is probably due to a part of the end resistance being measured in the pressure drops. From these curves it can readily be seen that there is no disturbance at the entrance due to the condensate being caught by the steam. The curves are those to be expected when considering the flow of steam in pipes where there are no peculiarities of action such as changing the direction of flow of a liquid in a pipe during one consideration.

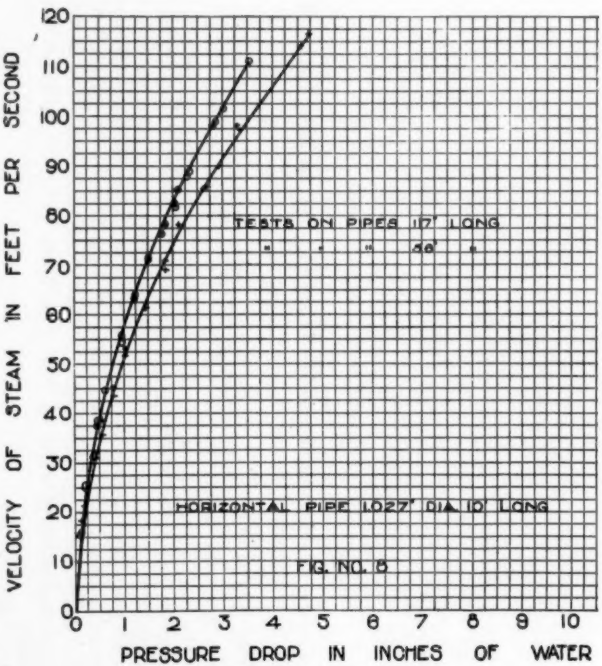


Fig. 8. Results of Tests (Class 2)

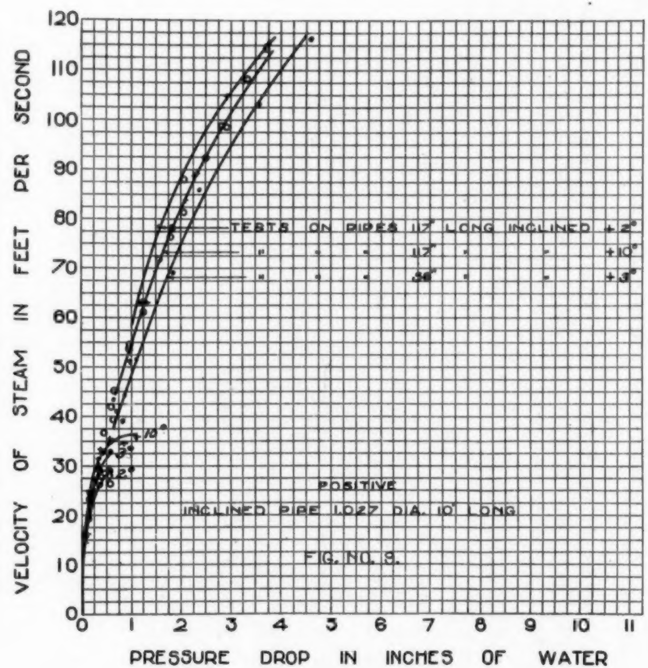


Fig. 9. Results of Tests (Class 1)

Class Three—Curves in Fig. 10.

The curves for the tests made on pipes with negative slopes show the same characteristics as the curves for horizontal pipes. The flow of the water with the steam

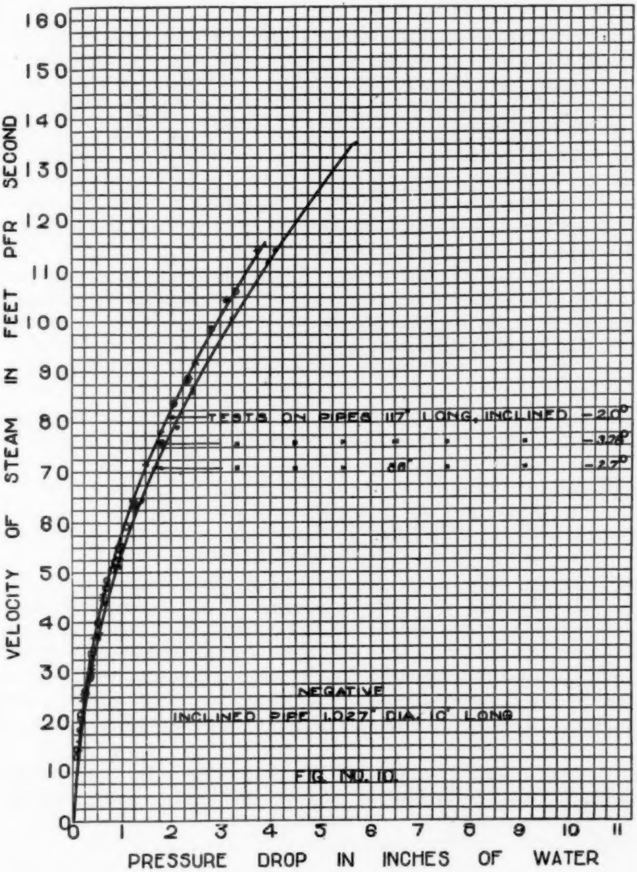


FIG. 10. RESULTS OF TESTS (CLASS 3)

does not affect the steam velocity; that is, there is no ejector effect of the condensate at the small slopes tested.

If the curves in Figs. 5, 8, 9 and 10 are compared, interesting conclusions may be drawn. These curves are from tests made with pipes of approximately the same diameter. If allowances are made for the corrections for lengths and variations in other variables the curves may fall together. In each of these figures the velocity of steam of one hundred feet per second gives a pressure drop of 3 in. The

upper portions of the curves are practically the same. In Figs. 8 and 10 the curves throughout are approximately equal and are continuous because there is no disturbance due to the condensate. In Fig. 9 where there is a slight counter flow of condensate there is a slight disturbance which becomes very pronounced as the slope is increased to 90 deg. as shown in Fig. 5.

From these curves it is evident that the position of the pipe in which the steam flows, affects the relation of the pressure drop to the velocity, when there is any moisture in the steam. In most house heating installations the steam is wet and the conditions under which this investigation was made, are those which are usually found in practice.

Similar test data should be obtained for all sizes of pipes in horizontal, vertical and inclined positions to determine the magnitude and location of the region of the critical velocity for each size of pipe. The resistance of any steam line could then be figured accurately because the data would be available to allow for the action of the condensate in vertical and inclined pipes.

As stated in the beginning of this report the investigation was interrupted before its completion, but sufficient data was obtained to realize the significance of the condensate in the supply line of a heating system, and the completion of this work would be valuable to the heating engineer.

No. 721

SOME FACTS ABOUT ENCLOSED RADIATION

By R. V. Frost, Norristown, Pa.

MEMBER

DIRECT radiation concealed by casings, known by the common term of *enclosed radiation* is becoming ever more popular among architects for any installation where interior decoration is a factor; but unfortunately, very little data are available to assist them and their engineers in properly designing the enclosures.

An attempt has been made by the Society, by the heating and piping contractors, and by several handbook publishers to gather together data on the subject, but owing to the lack of authentic data these several attempts are noteworthy more particularly for their remarkable lack of agreement than for their value to the profession. It is the purpose in this paper to point out several interesting phenomena in connection with enclosed radiation with the hope that the discussion thus aroused will stimulate a keener interest in their investigation.

As a general statement, the problem of enclosed radiation is one of air circulation. Air must be made to circulate with the greatest possible freedom over every part of the heating surface of the radiator. If there is a restriction, then there is a reduction in heating capacity. With the information available at present, it is impossible to tell, except in a general way, how this is to be done. It is known, however, that an enclosure can be designed for a radiator that will cause the radiator to emit more available heat than can be obtained from the same radiator when bare.

As stated previously, this Society has attempted to assist the profession by furnishing data on the subject, this data being presented in the Code under Section IV on Direct Steam and Hot Water Radiation. As there presented, the types of enclosures are divided into six different general classes and they are used here in the same grouping.

Referring to the accompanying illustrations: Case I has a solid panel in front of the radiator and is open at the top and bottom. It is a type that is very useful in schools, offices and churches, where it is desired to protect nearby persons from the direct heat of the radiator. It can be made fully 10 per cent more effective than a direct radiator.

In this type of enclosure there is but one thing to keep in mind. The dimension B must be made large enough. It is not known what the minimum size may be,

Presented at the Annual Meeting of the American Society of Heating and Ventilating Engineers, Boston, January, 1925.

but for safety the opening ought not to be less than 80 per cent of the free area of the outlet air passage.

Case II is practically the same as Case I except for the resistance offered by the grills. Some people are afraid of this type of enclosure, but it really is a very effective one, provided the free areas through the grills are made ample for free circulation. Many grills are so designed that the net air space is less than 50 per cent of the gross area of the grill. In such a grill trouble is sure to result unless the gross area allows for the restriction.

Case III is not radically different from Case II, but care must be taken that dimensions *D* and *C* are sufficient and that a smooth easy curve is provided on the back lining, so that the change in direction of air movement need not retard the velocity of air flow. As dimension *D* is increased, dimension *C* may be reduced until it equals dimension *A* of Case II. Dimension *D*, to give good results, should not be less than the width of the enclosure. This type of enclosure can be made equally as effective as a direct radiator and even more so on some installations. The writer has used this type on a number of occasions and always with excellent results.

Case IV is a very common type, especially for window radiators under a seat and is apt to be very much misused. A broad shelf over a radiator, if set close to the top of the radiator is about as effective as a blanket in stopping the emission of heat. If the shelf covers but a portion of the radiator, say one-half, the reduction in capacity is probably not more than 10 per cent, but if the shelf extends out beyond the front of the radiator, say from one and one-half to twice its width, then the reduction probably runs as high as 35 per cent.

Case V is parallel to Case IV, but the grill over the front of the radiator reduces the heating capacity in varying percentages in ratio to the reduction in free area. Thus a grill with but 50 per cent free area is much more detrimental than one with 80 per cent. What the percentage of reduction may be cannot be definitely told but it must be around 35 per cent, while an 80 per cent grill does not reduce the capacity more than 10 per cent. When the reducing effect caused by the grill is added to that of the shelf, then the combined effect must be as high in some cases as 50 per cent and perhaps more. An open grill over the full front of the radiator is not equivalent to the grills in Case III, because in Case III there is a chimney effect created which produces a more positive circulation of air over the heating surface.

Case VI is equivalent to a direct radiator except for the retarding effect caused by the resistance of the grills to free circulation. This ought not to exceed 5 per cent, but undoubtedly goes as high as 20 per cent, or even more when some types of grills are used.

An adaptation of Case VI, recently employed and described by W. H. Driscoll, chairman of the Committee on Research, is worthy of especial note, for it secures the advantages of Case I at a very moderate cost. In Mr. Driscoll's application, a close fitting sheet metal casing was placed about the radiator so that it rested upon the lower hubs and extended up to the upper grill. The air then entered through the front grill, passed down to the bottom of the radiator outside the casing, and then up over the heating surface of the radiator and out through the upper grill. This scheme obviates the construction of an expensive insulated lining and allows the architect a greater freedom in the design of the enclosures.

From the foregoing outline of the six cases it will be noted that the most important factors to be considered by the architect are to select grills that have a

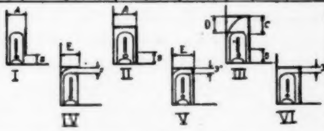
EXPERIMENTAL DATA ON ENCLOSED RADIATION

Date of Test	Radiator	Surface Sq. Ft	Enclosure	Length of Test Hrs	Outside Temp	Room Temp		First Hour			Last Hour			Total		
						Start	Final	Temp. Rise	Cond. Lbs.	Temp. Diff. Steam-Air	Temp. Rise	Cond. Lbs.	Temp. Diff. Steam-Air	Temp. Rise	Cond. Lbs.	Temp. Diff. Steam-Air
12	Peerless Scul. 14 sec. 26"	52½	Enclosed	5	62	68	94	17	10.25	132	1.5	9.36	123.	26.	48.5	635
			Bare			74	95	12	10.45	131	1.5	9.78	122.	21.	51.14	629.5
	Excell. Jr. 9 sec. 8	72	Enclosed			72.5	92	18.5	8.7	133	1.5	7.7	125	19.5	41.	643
			Bare			73	91.5	18.	9.26	133	1.5	8.73	125.5	18.5	44.5	644.5
13	Peerless	52½	E	7	48	61	95.5	24.5	10.48	131.5	0.5	8.87	121.5	34.5	67.38	876.5
			B			60	97.	26.	10.86	131.	0.5	9.66	120.	37.	71.10	870.
	Excell.	72	E			57	93.5	26.5	8.83	133.5	0.5	7.85	123.5	36.5	58.2	891.5
			B			57	94.	26.5	9.7	133.5	0.5	8.9	123.	37.	64.	889.5
14	Peerless	52½	E	7	52	59.5	98.5	27.5	10.11	130	1.0	8.69	118.5	39	65.26	864.5
			B			58.	96.	26.	11.08	138	1.0	9.71	121	38	72.1	833.5
	Excell.	72	E			60	94.	24.	8.15	139	1.5	7.82	122	34	57.4	891
			B			58.5	94.5	25.	9.54	133.5	1.5	8.46	122.5	36	62.1	892
16	Peerless	52½	E	6	64	70.5	99.5	19.	9.74	127.5	1.0	8.66	117.5	29	54.8	730
			E			69.5	98.	19.5	10.49	128.	1.0	9.43	119.	28.5	59.3	736.5
	Excell.	72	E			69.5	95	17.	8.34	130.5	1.0	7.40	122	26.5	46.9	753
			B			70.5	96	15.5	8.73	131.	1.0	8.55	121.	25.5	51.5	754.5

Enclosure E. Wood box ⅞" thick, lined ½" asbestos and bright tin. Open at top. 5" space at bottom. 35½" long. 29" high. 10" deep.

Enclosure B. Radiator bare, no enclosure.

Enclosure E'. Wood box ⅞" thick, lined as above. Open at top. 3" space at bottom. 44" long, 35" high, 12" deep.



free area sufficient for the condition and to design the deflectors so that the change in direction of the air flow can be effected with the least loss in velocity.

One other important point to be observed and that is but little considered, is the character of the lining. The lining is usually made heat resisting by the use of insulating material in order to prevent warping of the wood casing, but as a matter of fact the character of the metal surface is of greater importance than the insulation. If the metal lining has a polished reflecting surface, the effectiveness of the heating surface is appreciably increased. This point will be brought out in our discussion further on, but at present suffice to say that a bright tin is the most adaptable metal to use. A good quality of sheet tin if properly installed, is equal to galvanized iron in durability and superior to it in appearance. The tin should not be painted, for when used behind a grill it is practically invisible, appearing almost black.

The attempt has not been made, either in the Code or in this paper, to give definite valuations for any of the cases, for these valuations are too largely affected by the individual design. Just to show the degree of variation in these percentages, two authorities are quoted and compared with the Code; one, the Book of Standards published by the *Heating and Piping Contractors National Association*, the other, a trade data book.

Case I is not given at all by the heating and piping contractors, while the data book gives a 100 per cent value as against 110 per cent maximum in the Code.

In Case II, the heating and piping contractors add 10 per cent to the radiation, the data book adds 30 per cent, while the Code states that a safe value is 105 per cent, if the enclosure is properly designed.

In Case III, the heating and piping contractors add 20 per cent to the radiation, the data book adds 35 per cent, and the Code gives a valuation of 100 per cent, if the enclosure is carefully designed. The Code is upheld in this valuation by one manufacturer of a casing of this type, who advertises that his radiator cover makes a radiator slightly more effective than when direct.

In Case IV, the heating and piping contractors add 5 per cent for high radiators and 25 per cent for window radiators. The data book advises adding from 10 per cent to 15 per cent for high radiators, while the Code adds from 10 per cent to 35 per cent from high to low.

In Case V, the heating and piping contractors add from 10 per cent to 20 per cent for high radiators and 40 per cent for low ones. The data book adds 25 per cent, while the Code adds 30 per cent as an average.

In Case VI, the heating and piping contractors advise adding 10 per cent, the data book adds 20 per cent, while the Code adds but 5 per cent, provided the grills are not restricted.

A few years ago the writer was engaged in carrying on some experimental work along this line and the results were so unusual and unexpected as to be worthy of especial note. The work was performed at the Institute of Thermal Research of the American Radiator Co., Buffalo, N. Y., the purpose being to furnish data to an architect who believed that the indirect type of radiation would be more effective and economical when enclosed than would a column type of radiation.

The data from a few representative tests is recorded for the purpose of illustration in the table here presented. These tests were the final of a series covering more than three months, there being nearly 40 tests in the full series. The figures given cover four consecutive days' tests and show quite clearly the very interesting facts learned.

Conditions of Tests

For the purpose of making these tests a group of four rooms were used, of equal size and having exposures both inside and out of the building. Uniform temperatures were maintained in all adjoining rooms, on all sides, and above and below. The outside exposure was to the north toward an open field. The walls were brick and the windows all of the same size. The rooms were about ten by twelve in size, with nine foot ceilings.

The radiators used were two two-column direct type of equal size and two fin type indirect radiators of equal size. The radiators selected were chosen because their space dimensions are about equal. Two enclosures were provided, one for each of the two types of radiators tested, the enclosures being identical in every detail.

In making the tests, one radiator was set in each of the four rooms and the condensation and temperature data from each room recorded. Since the exposure in all four rooms was equal, the room temperatures and condensation weights were the only variables.

The data for a column radiator enclosed could then be compared with that for the same radiator bare, likewise, the indirect radiators could be compared, and then the column radiators both enclosed and bare could be compared with the indirect radiators both enclosed and bare.

To make the tests uniform, the radiators were interchanged each day so that the

same radiator did not occupy the same room twice, and the radiators were set against an interior wall in order to guard against excessive loss to the outside.

Before starting a test all the radiators were heated to steam temperature, maintained uniformly at 217 deg. The windows of all the rooms were opened during this period so that the rooms would reach as low a temperature as possible. Then at a given signal all the windows were closed, the room temperatures recorded and an automatic weighing device connected in. The tests were continued for from five to seven hours, the temperatures and condensation weights being recorded at stated intervals, so that each hour's run was, for all practical purposes, a complete test. The condensation weights were recorded in one-hundredths of a pound and the temperatures in one-half degrees.

It will be observed in the table that the column radiation is rated at 52.5 sq. ft. while the indirect of equal space dimensions is rated at 72 sq. ft., this including the extended surface.

Record of Tests Interesting

In the record of the tests one observes that on the 12th it was impossible to bring the room temperatures at the start of the test to as uniform a condition as desired. But it is worth while noting that at the end of the test, the room temperatures on the column radiators were very close together, and similarly on the indirects. It will be observed that this same condition prevails through all the tests on the four days and that this condition of uniform temperature is reached at the end of the first hour. Why this should be is one of the interesting questions. It is one of the unusual phenomena in these tests for which there must be some reason that is not evident at the present time.

Under temperature rise it is worth while noting that over 60 per cent of the rise occurred within the first hour. But also note that the condensation for the first hour is not more than 15 per cent greater than that for the last hour. We have often heard that it is much easier to warm fresh air than stale air, but a better explanation of this would be that it took about a 25 deg. temperature differential between the room temperature and outside air before the point of balance could be reached between heat transmission from the room and heat emission from the radiator, and that during the first hour all the heat given off by the radiator went to heat the air in the room.

Next observe that during the first hour, and also during the total time of the test, the temperature rise with but two exceptions was higher in the room containing the radiator enclosed than in the room containing the same radiator bare; the exception being on the 13th in the case of the column radiator and on the 14th in the case of the indirect. But also note that the condensation was in every test higher for the bare radiator as compared to the same radiator enclosed, the percentage of increase being approximately 10 per cent. Were it not for this observation, it would at once be concluded that the bare radiator has about 10 per cent greater heating capacity than the enclosed, but here we are confronted with the fact that the enclosed radiator although condensing less steam, has in the majority of cases given a higher temperature rise to the room. This is a most important phenomenon and one that should be given the most thorough scrutiny.

Some light may be given by the fact that this condition could be produced only by the use of a bright tin lining for the enclosure. Galvanized linings, black iron linings and paints of various kinds were used, but tin seemed to be the only thing that would turn the trick. An explanation offered is that the reflecting surface

of the tin lining stopped the flow of radiant heat from the radiator and threw it back to be more effectively taken up by the air in convection and so given out as sensible heat to the air in the room. It is for this reason that the enclosures of the type of Cases I, II, and III are given higher effective values than Cases IV, V, and VI.

A further point worth noting is that enclosure *E* was rather tight fitting to the radiation, there being but one-half inch excess space between the radiator and the casing at the front and back and practically no excess in the length. On the 16th this casing was compared with one having more space all around (marked *E'*) and found that the larger casing did not seem any more effective than did the bare radiators on the previous day's tests. This would indicate that the casings should be tight fitting to the radiators.

As a final observation, it is interesting to note that while the indirect radiator enclosed showed on but one occasion a temperature rise equivalent to the column radiator enclosed, the column radiator in every case showed a condensation rate from 15 to 20 per cent greater than the indirect,—a saving worth thinking about.

Summarizing this subject, one is impressed that the present method of testing radiation by which only the condensation is measured is very incomplete, and that a new method must be developed,—a method that will cover all three modes of heat transfer,—radiation, conduction and convection, if all the data necessary to properly study the subject of heat transfer from radiation is to be secured.

Dean Allen was engaged upon this subject when his untimely death cut short its completion. Since then there have been no published reports on the subject, although there has been expressed from time to time a very emphatic sentiment for an improved method.

Obviously any improved method must employ an insulated room or chamber in which all the energy emitted both above and below as well as within the range of sensible heat could be measured.

DISCUSSION

PRESIDENT ADDAMS: This fine paper which has been under preparation for a period of years is now open for discussion.

H. R. LINN: I would like to ask the speaker if he tried putting the water pan above.

S. A. JELLETT: The Peerless radiator was 26 in. high, presumably prime surface; the indirect radiator is not given. What I want to get is the flue velocities for the enclosed radiator. Many years ago the first flue radiator put on the market was an extended surface radiator. A few years later the prime surface radiator was produced. There was a marked increase in the velocity of the air through the flues in the prime surface radiator. We made a number of tests and found that the 26 in. radiator was in proportion to the surface very much more effective than the 38 in. radiator. In this case the indirect radiator is evidently 36 to 38 in. high, while the other is only 26 and I presume there is extended surface on the indirect resulting in less effective heat emission to the surrounding air which affects the velocity of the air in the chamber.

It seems to me we should have data by which we could determine the relative

efficiency of the surface warming the moving air. I don't know whether Mr. Frost in his test has gone far enough to measure the difference in velocity of the air moving in the flues. It has been my experience that when you increase the height of the radiator you decrease the effectiveness of the flues, because the velocity is not increasing with the height of the radiator.

As a result of our own tests we decided wherever possible not to make them more than 26 in. high, because we get better results per square foot of surface and per dollar than with the higher radiator.

R. C. BOLSINGER: I would like to say something regarding a standard method of testing radiators. I think the Society should adopt a code and some standard for this work. It is the plan to publish in the new minimum requirements code the set of radiation tables compiled by the Research Laboratory. I don't believe that that standard has ever been accepted. Until the Society adopts that as a standard, I don't see how it will be possible to include the table in the code.

R. V. FROST: In answer to Mr. Jellett, the radiator that we used, the Excelsior, was an 8 ft. section, 26 in. in length and the section stood vertical so that it was in practically the same position as a column radiator. In that case the fins were across the air flow. This was not a flue; it was simply the extended surface extending into the flue passage and the heat was taken off from the ends of the fins instead of being in active contact with the prime surface of the radiator. You will note that the enclosure used most of the time was only 29 in. high and we allowed some space above.

The reason why we took the Excelsior and why we used the 26 in. was because those were the two radiators of the column type and the indirect type that were approximately the same in all dimensions. They were the nearest we could get. We didn't attempt to measure the air flow. The only things we were looking for were the heat output and the condensation.

R. C. BOLSINGER: Several years ago an engineer conducted some experimenting to determine the proper location of a radiator in a room. He had two rooms of the same dimensions and the same exposures; he placed the radiators in different locations in each room. He ran the two tests together and determined the time required to raise the temperature in these rooms seven degrees. In one case the condensation was 25 per cent greater than the other.

E. H. LOCKWOOD: (Written discussion.) A natural explanation can be made for the disproportionate rise of room temperature during the first hour of the run, based on the procedure adopted in starting these tests. It is stated that the room was cooled off in the morning by opening the windows. This kind of cooling would serve to cool the air but probably not the walls and furniture, hence the rapid air temperature rise when the windows were closed at start of test.

The author points out another phenomenon which is a real heating paradox, namely, less steam condensed in certain cases yet more heat given to the room. The same paradox has been reported by at least one other experimenter, also the suggestion that radiators be tested for ability to heat the room, not the ability to condense steam.

I believe it can be safely stated that there is actually no such phenomenon when a radiator is properly tested, and that the weight of steam condensed is a true measure of ability to heat the room. The proper method of testing referred to is to prevent the escape of radiant heat through the walls. This radiant heat loss occurs prin-

cipally at the wall behind the radiator, which should be screened by a thin shield with air space. Other radiant heat loss occurs from outside walls facing the radiator, which can be prevented by suitable screens. Hundreds of radiator tests at all seasons of the year have convinced me that the weight of steam condensed is a satisfactory and accurate measure of radiator performance.

No. 722

BASING WARM AIR HEATER SELECTION ON CLIMATOLOGICAL CONDITION AND HEATER PERFORMANCE CURVES

V. S. DAY, URBANA, ILL.

MEMBER

A PROPER selection of size and type of warm air heater implies fulfillment of the following requirements:

a. The heater must operate at maximum efficiency throughout major portion of heating season.

b. The heater must have sufficient stand-by capacity to carry extreme weather condition loads with safe temperatures and without damage.

c. The heater must have capacity to carry the load for which it is designed with reasonable firing intervals, fuel reserve for ignition of fresh charge, and with a reasonably moderate register air temperature.

d. The heater must be capable of performing with good efficiency at the very low rates of combustion required in mild weather.

A heater capable of meeting all of these requirements must be thoroughly tested and its performance characteristics known for a wide range of combustion rates varying from the smoldering fire to the maximum rate obtainable with the draft of a good chimney. If the performances are known for several heaters for such a range of combustion rates, one may be selected which will satisfy the above requirements better than any other.

A furnace which will satisfy these requirements in one locality, or under one set of climatological conditions may be unsatisfactory in another. It is essential, therefore, that the climatological data for each locality be known. A study of the Weather Bureau records uncovers some very useful data. For example, an analysis of the temperature records for the district of Chicago for a period of 48 years, and the resultant effect on furnace performance, is given.

In Fig. 1 the average annual temperature record (for Chicago) is shown graphically. Three curves are shown, the mean, the mean maximum, and the mean minimum. The mean temperature passes below 60 deg. fahr. about the first of October and remains below 60 deg. fahr. until the latter part of May. It is generally agreed that heating is necessary when the temperature falls below 60 deg. fahr. On this basis the heating season at Chicago is slightly less than eight months

Presented at the Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Boston, January, 1925.

TABLE 1. CONDENSED HEATING SEASON DATA FOR THE UNITED STATES

	Approx. Number Days Below 60° Fahr.	Mean for Heating Season ° Fahr.	Mean for Coldest Month ° Fahr.	Coldest of Record ° Fahr.	Ratio of Heating Loads: Coldest month to average heating season	Ratio of Heating Loads: Coldest liability to average heating season
Alabama, Birmingham	150	50	45	—10	1.20	4.00
Arizona, Kingman	181	51	44	9	1.48	3.14
Arkansas, Little Rock	151	47	42	—12	1.22	3.56
California, Sacramento	181	52	46	19	1.33	2.84
Colorado, Denver	243	42	31	—29	1.39	3.54
Connecticut, New Haven	243	40	27	—14	1.43	2.80
Delaware, Dover	212	43	35	— 5	1.35	2.88
Florida, Jacksonville	90	56	56	10	1.03	4.30
Georgia, Augusta	151	51	47	— 8	1.21	4.10
Idaho, St. Maries	273	42	30	—26	1.43	3.42
Illinois, Springfield	212	40	27	—24	1.43	3.14
Indiana, Indianapolis	212	40	29	—25	1.37	3.16
Iowa, Sioux City	243	36	19	—35	1.50	3.08
Kansas, Concordia	212	40	28	—25	1.40	3.17
Kentucky, Bowling Green	212	46	37	—20	1.37	3.75
Louisiana, Shreveport	151	52	47	— 5	1.28	4.16
Maine, Portland	273	39	22	—17	1.55	2.80
Maryland, Baltimore	212	43	34	—15	1.33	3.15
Massachusetts, Boston	243	39	27	—13	1.39	2.68
Michigan, Detroit	243	38	24	—24	1.44	2.93
Minnesota, St. Paul	243	32	12	—41	1.53	2.90
Mississippi, Vicksburg	151	53	48	— 1	1.29	4.20
Missouri, St. Louis	212	43	31	—22	1.45	3.41
Montana, Helena	302	39	20	—42	1.61	3.62
Nebraska Omaha	212	36	20	—32	1.47	3.00
Nevada, Winnemucca	273	42	29	—28	1.46	3.50
New Hampshire, Concord	273	38	21	—35	1.53	3.28
New Jersey, Trenton	212	42	31	—13	1.39	2.96
New Mexico, Santa Fe	243	40	28	—13	1.40	2.76
New York, Albany	243	38	23	—24	1.47	2.94
North Carolina, Raleigh	182	48	42	— 2	1.27	3.27
North Dakota, Bismark	273	32	20	—45	1.32	3.02
Ohio, Columbus	212	40	29	—20	1.37	3.00
Oklahoma, Oklahoma City	182	46	37	—17	1.37	3.62
Oregon, Portland	243	47	39	— 2	1.35	3.13
Pennsylvania, Harrisburg	212	40	29	—14	1.37	2.80
Rhode Island, Providence	243	40	27	— 9	1.43	2.63
South Carolina, Columbia	150	50	46	— 2	1.20	3.60
South Dakota, Pierre	245	35	17	—40	1.51	3.15
Tennessee, Knoxville	212	47	38	—16	1.39	3.74
Texas, El Paso	151	50	45	— 5	1.25	3.75
Utah, Salt Lake City	243	42	29	—20	1.46	3.22
Vermont, Northfield	273	34	15	—32	1.53	2.83
Virginia, Wytheville	212	42	34	— 8	1.29	2.78
Washington, Seattle	303	49	40	3	1.43	3.20
West Virginia, Parkersburg	212	42	33	—27	1.32	3.46
Wisconsin, La Crosse	243	35	15	—43	1.57	3.23
Wyoming, Sheridan	273	36	19	—45	1.50	3.38
				AVERAGE	1.39	3.27

duration. That the length of the heating season and the cold intensity varies widely for different localities may be seen in Table 1. Fig. 1 shows the average daily variation in temperature at Chicago to be about 13 deg. fahr.

The application of these temperature data to the furnace selection problem is continued in Fig. 2. Here a residence heating requirement for the heating season is shown graphically. The heat loss[1] is calculated for 0 degree weather and increased 20 per cent to account for transmission losses between the heater and the room. No special account was taken of the effect of wind upon heat loss in calculating the heat loss for the building. The ordinates in Fig. 2 represent, therefore, the hourly B.t.u. capacity required of the heater. The average heater capacity for the season is shown to be less than half of the capacity required for the zero weather, two-thirds of the capacity required for the coldest month, and about one-third of the

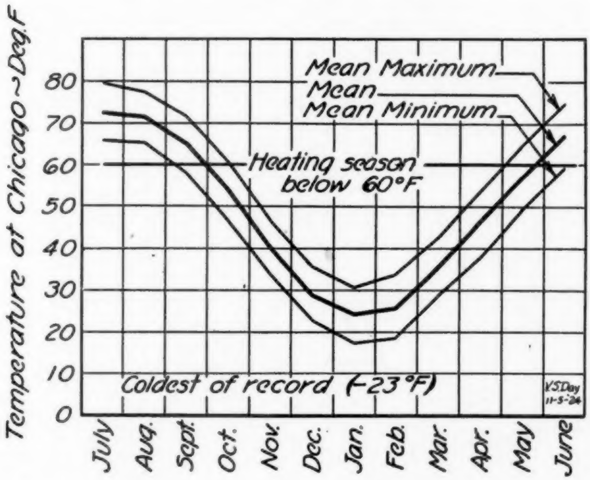


FIG. 1. AVERAGE AND EXTREME TEMPERATURE DATA FOR THE CITY OF CHICAGO FOR PERIOD OF FORTY-EIGHT YEARS

capacity required for the lowest temperature of record (—23 deg. fahr.). The daily capacity range amounts to about 30,000 B.t.u. per hour or one-fifth of the zero weather capacity. If, from a set of performance curves, Fig. 3, obtained by tests of a typical furnace, the combustion rates and efficiencies of the heater corresponding to the capacity curves of Fig. 2 are plotted, the curves of Fig. 4 and Fig. 5 result.

Fig. 3 is a performance record[2] for two heaters having the same grate diameters. The capacity in B.t.u. per hour and the efficiency, are shown for each for the range of combustion rates available with a very good chimney.

By plotting combustion rate values from Fig. 3, heater A, corresponding to the capacities of Fig. 2, the curves in Fig. 4 were obtained. The diverse conditions

[1] Calculations based on the Warm Air Heating Research Residence of the *National Warm Air Heating and Ventilating Association* at the University of Illinois.
[2] See Bulletin No. 141, Engineering Experiment Station, University of Illinois, for performance curves from actual tests.

under which a heating plant must function is emphatically shown in this diagram. The average heating season combustion rate for this particular furnace installation is only 3.4 lb. coal burned per sq. ft. of grate per hour. In zero weather, however,

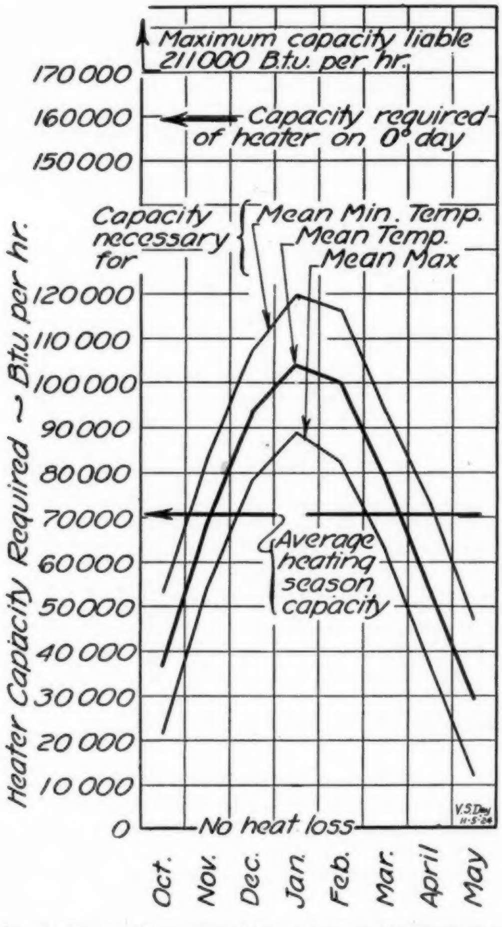


FIG. 2. HEATER CAPACITIES NECESSARY FOR MAINTENANCE OF A TEMPERATURE OF 70° F. IN A RESIDENCE BUILDING IN CHICAGO

the plant must operate at a combustion rate of 8.4 lb. per sq. ft. grate per hour, and to insure heating under the (—23) deg. fahr. weather shown by record to be liable in Chicago, heater *A* must operate at a combustion rate of 12.7 lb. per sq. ft. per hour. The peak load on the chimney and grates may be four times the

average heating season load. The average daily range of combustion rate is represented by the ordinate between the mean maximum and mean minimum lines. To function economically under such a wide range of conditions this furnace should have an ideally high and flat efficiency curve with the highest point on the curve

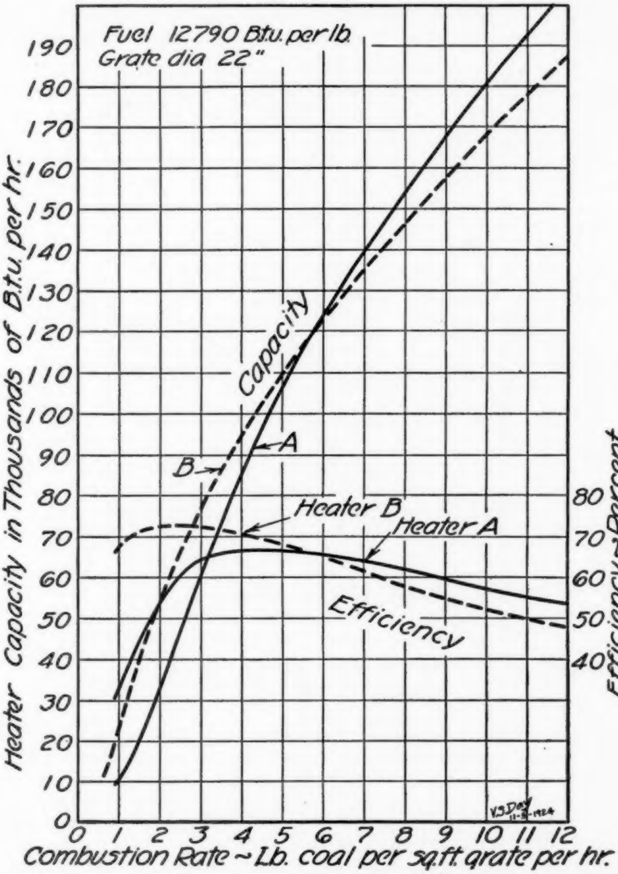


FIG. 3. PERFORMANCE CURVES FOR TWO WARM AIR HEATERS

at the combustion rate corresponding to the average heating season combustion rate. The efficiencies of the heaters represented by Fig. 3 for the season are shown in Fig. 5.

Of the two heaters represented, that having the higher efficiency curve at low rates of combustion is shown to be the most efficient throughout the heating season and would be the better selection for such temperature conditions as are found at

Chicago. This may be seen in Fig. 5 in which the seasonal efficiency for heater *B* is higher than for heater *A*, although during extremely cold weather heater *A* operates at higher efficiency than heater *B*.

Such direct comparisons of seasonal efficiency and performance are only possible between heaters of the same grate size. If both grate size and performance characteristics are different, the relative economy of two heaters may be compared by finding their seasonal efficiencies from performance curves similar to those of Fig. 3,

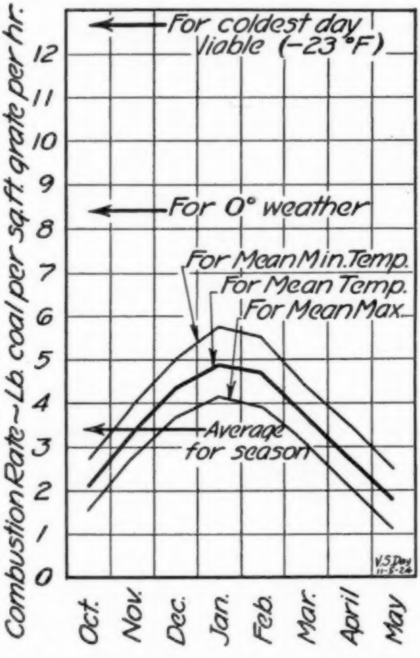


FIG. 4. COMBUSTION RATES FOR THE HEATER *A* CORRESPONDING TO THE HEAT CAPACITIES SHOWN IN FIG. 2

but plotted on a basis of "total hourly fuel consumption" rather than "combustion rates in lb. of fuel burned per sq. ft. of grate per hour."

The engineer should be familiar with the performance curves for the various sizes of each line of heaters, plotted on the basis of "total hourly fuel consumption," as well as the performance curves of common sizes of different lines of heaters plotted on the basis of "combustion rate."

That climatological temperature conditions are of much importance in the selection of furnace sizes and types is evident from Table 1, in which a wide variation in heating season mean temperatures, coldest month temperatures, and coldest liability temperatures for various sections of the United States are shown. The data are by no means complete, as special conditions such as altitude, mountain

ranges, and large bodies of water, have a great effect on the temperatures of relatively small districts. The data given are representative of the whole of the states rather than small divisions of the states.

In Column 2 of the table the heating season is shown to vary from three to ten months. In those localities in which the season is short the mean temperature for the heating season as well as the temperatures for the coldest month and the coldest day are shown to be relatively high. Small sizes of heaters having arched efficiency curves could be used with good economy in such localities.

In Column 3 of the table the average mean temperature for the heating season is shown. The heater selected for a particular residence must have its maximum efficiency at the capacity corresponding to the heat loss calculated from this mean temperature. It must also be capable of efficiently carrying the load corresponding to the coldest month. Column 6 shows that the load for the coldest month

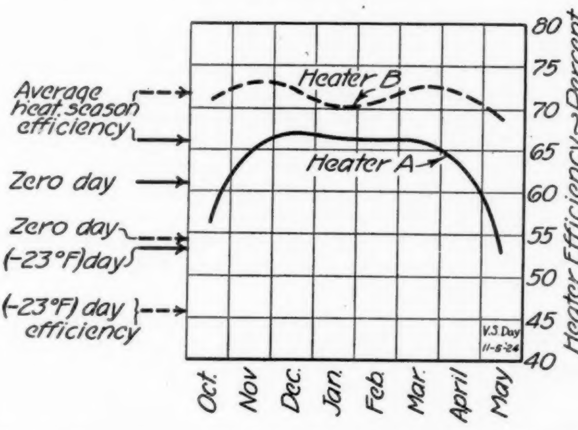


FIG. 5. COMPARISON OF SEASONAL EFFICIENCIES FOR TWO HEATERS REPRESENTED BY PERFORMANCE CURVES IN FIG. 3

may be slightly greater than for the entire season in some localities (in the southern states 25 per cent greater) but much greater in other localities (50 per cent in the northern and western states). In those localities in which the ratio of the load for the coldest month is only 25 per cent greater than for the average load for the heating season, heaters having arched efficiency curves may be employed without appreciable lack of economy, but if the ratio of the January load to the average seasonal load is 1.50 or 1.60, only heaters having flat efficiency curves should be used.

In Columns 5 and 7 of the table the extreme temperature data are given. Column 7 shows that no furnace should be selected for a particular installation unless it is capable of developing a capacity 2.75 times the capacity at which it will normally operate. In some localities the furnace must be capable of developing a capacity four times as great as the normal operating capacity.

For another reason no furnace should be used unless it is capable of developing a very large standby capacity. This is the necessity for a large reserve capacity

during periods of rapidly decreasing outdoor temperature, or during early morning firing. During early morning firing the combustion rate frequently is limited only by the chimney draft, and may even exceed the rate for the extreme temperatures mentioned in Table 1.

Good average ratios from Columns 6 and 7 of Table 1 are shown to be 1.39 for the ratio of the heating load for the coldest month to the heating load for the average of the heating season, and 3.27 for the ratio between extreme and normal load weather conditions. The two values should be useful in the design and selection of heaters.

Tests show considerable variation in the curves of efficiency and capacity for different heaters, and the importance of knowing performance characteristics of heaters of all kinds is becoming more appreciated by engineers.

DISCUSSION

S. A. JELLETT: There is only one suggestion I have in regard to Table 1. I think we ought to state the locality in the state for which the temperature is given for the Table gives a value far from being the lowest temperature on record in the State of Pennsylvania, for example. Possibly it is correct for the city of Philadelphia. It is not correct, however, for Scranton or a number of other important cities in the State.

H. M. HART: I don't want to discuss the paper; I only want to express my appreciation for the careful study that has been given and the fact that our attention is called to something that we sometimes overlook and that is the point that Mr. Willard stressed in his presentation of the paper regarding fuel consumption rates and the rate of combustion.

A. C. WILLARD: The suggestion that specific localities be named in Table 1 is a very pertinent suggestion and I am in favor of it for several reasons. Thank you for the suggestion.

No. 723

CHARACTERISTICS OF AN AIR-TUBE TYPE COPPER HEATER

By L'Roche G. Bousquet (Non-Member) and George A. Foisy (Member)

LOWELL, MASS.

THE air-tube cellular type of copper heater, the characteristics of which are to be presented in this paper, differs radically from heaters ordinarily used in indirect heating systems. In appearance it looks like an automobile radiator of the honey-comb type. It is extremely light in weight. It is compact, offering an exceptionally large amount of heating surface for a given space occupied. For velocities of 900 ft. per minute and above, this type of heater will dissipate more B.t.u.'s per hour than other types of heaters with a given frictional loss and volume of air handled.

Scope

The object of this paper is to furnish to the heating profession sufficient data on such heaters for their most efficient utilization in indirect heating systems. The data presented include curves showing:

1. Frictional resistance.
2. Rate of heat transfer.
3. Rate of condensation of steam.
4. Temperature rise of air.

In addition, tables showing in detail, final temperature and condensation per square foot per hour, are given for a wide range of entering air temperature and for seven different velocities.

Description of Heater

Because the use of the air-tube type of heaters for indirect heating is comparatively new, a description of its construction may be helpful. Fig. 1 shows a detailed drawing of such a heater. It is essentially a copper heater made up of seamless copper tubes bonded together with a special alloy. The steam is admitted to the core by means of a brass tank supplied with pipe fittings. The condensation is taken care of in a similar manner.

Apparatus Used

The wind tunnel used in testing is shown in Figs. 2, 3 and 4. The design of

---

Presented at the Annual Meeting of the American Society of Heating and Ventilating Engineers, Boston, January, 1925.

the wind tunnel is not original with the authors. Advantage was taken of the experience of the Bureau of Standards obtained in the building and use of an 8 in. square wind tunnel. The tunnel used in these experiments is 1 sq. ft. in cross-section so that the frontal area of the heater being tested, exclusive of the headers, is exactly 1 sq. ft. The tunnel is $21^1/_2$ ft. long, the core being inserted at a distance of 11 ft. from the inlet end of the tunnel. The inlet end of the tunnel is movable so that heaters of different depths can be tested. An absolute air-tight seal is obtained when the heater is clamped in place.

Every known precaution was taken in building the wind tunnel to attain as nearly as possible parallel air flow. In the first place, the fan draws the air through

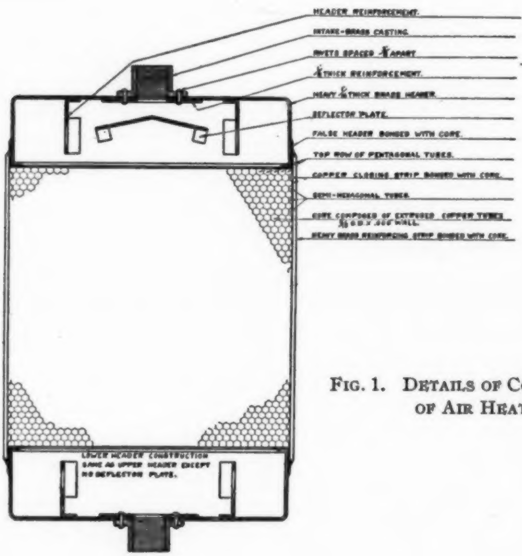


FIG. 1. DETAILS OF CONSTRUCTION OF AIR HEATER

the tunnel instead of forcing it through. The fan used is a Sturtevant of the multivane type. The velocity of the air is varied by changing the speed of a direct current variable speed motor. The air enters the tunnel through a bell-shaped inlet, fitted with a wind straightener. In inserting the pitot tubes, piezometer rings, and thermometers, care was taken to keep the inside surface of the tunnel absolutely smooth to minimize eddy currents.

The wind tunnel is supplied with two means of measuring dynamic head. One is an A. B. C. pitot tube attached in such a manner that it can be moved both vertically and horizontally. The other is a pitot grid, having sixteen dynamic openings all connected to one reservoir. The purpose of this pitot grid is to obtain an integration of the dynamic head. The static pressure, both in front and in back of the heater, is measured with piezometer rings, each set at a distance of 6 in. from the heater. Both inlet and outlet air temperatures are measured with Leeds and Northrup resistance thermometers. These thermometers are designed

in such a manner so as to give an average temperature of the air in the tunnel. This is accomplished by having six individual thermometers 12 in. long and spaced 2 in. apart, all connected in series. The two sets of thermometers are connected to one instrument with a double-throw switch. In this way the average inlet and outlet air temperatures can be measured almost instantaneously. The frictional loss through the heater is measured by connecting the piezometer rings in front and in back of the heater to a monometer tube. The static pressure on the inlet side of the wind tunnel, as obtained with the piezometer ring, in conjunction with a Fortin type barometer is used in determining the barometric pressure of the air in the tunnel. The humidity of the air is determined by two mercurial thermometers inserted in the wind tunnel. These thermometers are such as used in a Taylor sling psychrometer.

The steam supply is taken from a 2 in. line, trapped to remove excessive condensation, and reduced to a $^3/_4$ in. pipe to the heater. By the use of two needle valves in the line it is possible to obtain some superheat in the steam, the object of this being to prevent any condensation present in the steam line from being collected as part of the steam condensed by the heater. The condensate is collected in a covered

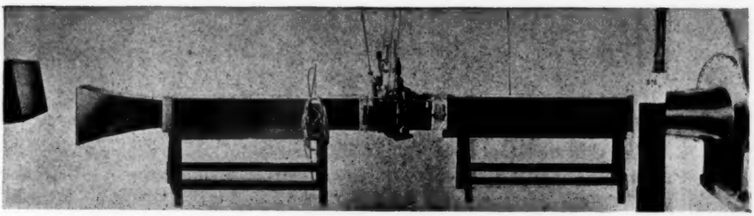


FIG. 2. WIND TUNNEL USED IN TESTING

galvanized can through a *U*-tube provided with a gage glass. The condensation level in the gage glass is controlled by a small needle valve at the outlet. The condensation is weighed on a small National scale which affords an accuracy of about 0.1 per cent in the range used. Attached to this scale is an electric signal which indicates when the beam crosses the half-way mark. A stop-watch is used to determine the period of condensation. The steam pressure is measured with a Schaeffer and Budenberg test gage with a range of from 0 to 20 lb., reading in pounds and ounces. The steam temperatures, both inlet and outlet, are measured with accurate mercurial thermometers inserted in the steam line by means of a hollow, threaded, copper plug screwed into a *T*. The outside of the plug is in contact with the steam, the inside being filled with mercury forms a receptacle for the thermometers.

The heaters used in the tests are shown in Fig. 5.

Experimental Work

1. *Calibration of the Wind Tunnel: a.* A traverse of the wind tunnel was made with an A. B. C. pitot tube. The details of this work are given in Appendix I. The coefficient obtained for use in calculating the average veloci-

ties through the tunnel is given in the formula:

$$V = 1096.2\, C \sqrt{\frac{p}{D}} \qquad (1)$$

V = Velocity feet per minute
p = Velocity head in inches of water
D = Density of air pounds per cubic feet
C = Coefficient obtained by exploration = 0.9396.

A comparison of velocity was made as obtained by the dynamic head with the A. B. C. pitot tube and with the pitot grid. As a result of this work the formula for use with this wind tunnel, for calculating velocity using the pitot grid for measuring dynamic head, and the piezometer rings for measuring static pressure, was found as follows:

$$V = 1040 \sqrt{\frac{p}{D}} \qquad (2)$$

As a further precaution to insure the accurate determination of velocities, the coefficient for use with the above expression was obtained by means of thermal measurements. This coefficient agreed with the original one to 0.79 per cent.

b. The steam thermometers were calibrated by comparing the temperatures indicated at 2, 5, 10 and 15 lb. steam pressure with the theoretical.

c. The resistance thermometers in the wind tunnel were calibrated before being installed. This was done by immersing the thermometers in a paraffin bath and comparing the temperatures with those indicated by a mercurial thermometer of known accuracy.

d. The three manometers used were compared with one from the American Blower Co. furnished with a calibration chart.

2. Method of Making Runs: Equilibrium was first obtained by running the apparatus at a definite air velocity and steam pressure for at least 30 min. Precautions were taken that the heater was free of trapped air. Twenty to 40 lb. of condensate was collected in each run in increments of 10 lb. Each run represented an average of from two to three determinations depending on the rate of condensation. One operator manipulated the steam valve, recorded the steam pressure, temperatures, and rate of condensation, while a second operator recorded the air temperatures, velocity and static heads, frictional loss, humidity and barometric pressure.

3. Results: Frictional losses in inches of water are represented in Fig. 6. The same data is plotted on logarithmic paper and shown in Fig. 10. From this latter graph equations have been derived that give the relation between velocity and frictional loss for different lengths of tubes in the heaters.

a. $F.\,L. = 0.0^{6}3763\ V_1^{1.825}$ (6 in. tube) .... (3)
b. $F.\,L. = 0.0^{6}346\ V_1^{1.825}$ (5 in. tube)
c. $F.\,L. = 0.0^{6}307\ V_1^{1.825}$ ($3^7/_8$ in. tube)
d. $F.\,L. = 0.0^{6}2602\ V_1^{1.825}$ (3 in. tube)
$F.\,L.$ = Frictional loss in inches of water
V_1 = Velocity feet per min. through heater.

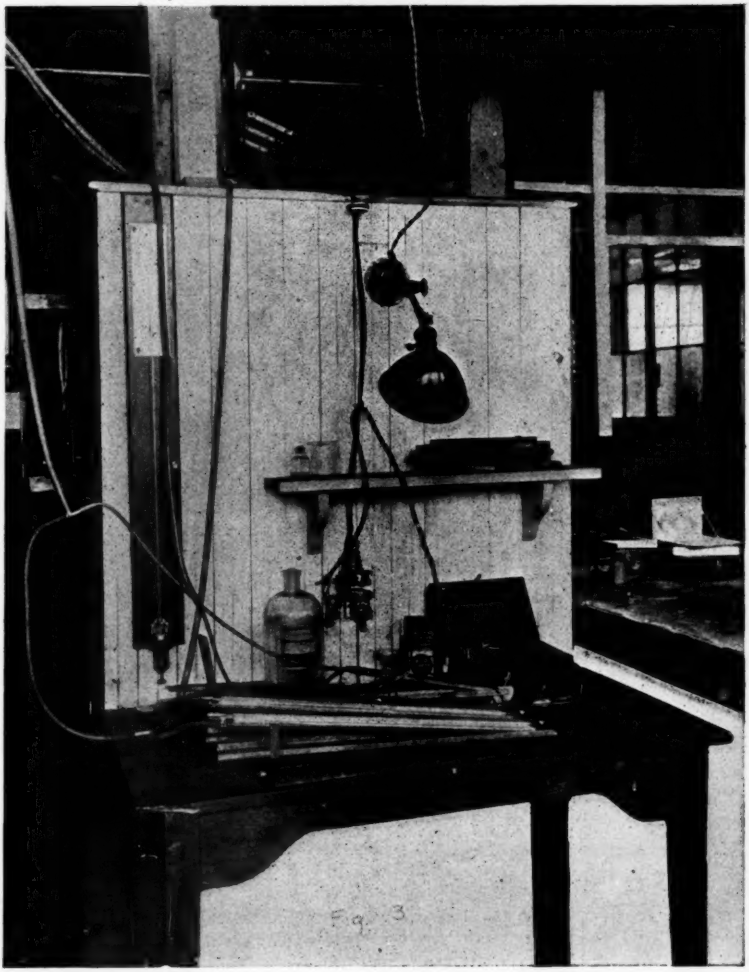


Fig. 3. View of Wind Tunnel

In Fig. 7 the relation between velocity and K (the heat transfer coefficient) is given. K was calculated as follows:

$$K = \frac{\text{Cond. per hr.} \times 960}{S \times \left(t_3 - \dfrac{(t_2 + t_1)}{2}\right)} \qquad (4)$$

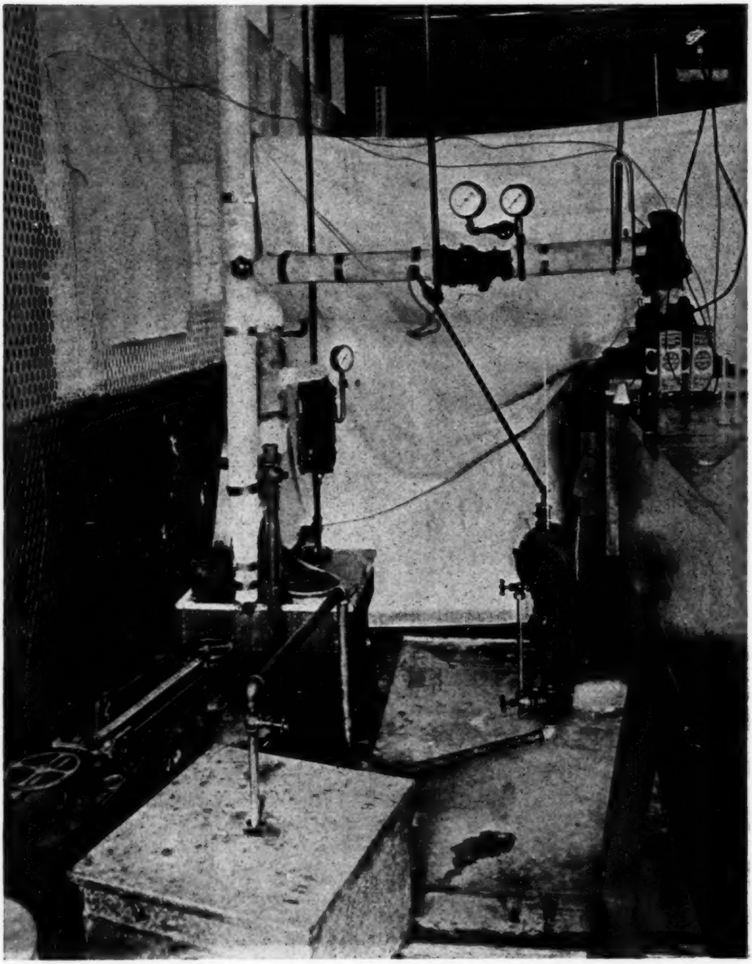


FIG. 4. VIEW OF WIND TUNNEL

K = B.t.u.'s per hr. per sq. ft. per deg. fahr. temperature difference
S = Net radiating surface in sq. ft.
t_1 = Temperature of air entering heater
t_2 = Temperature of air leaving heater
t_3 = 227.15 deg. fahr. steam temp. 5 lb. gage

Final temperatures of air leaving the heater for entering air at 0 deg. fahr. are shown in Fig. 8, while condensation per square foot per hour is shown in Fig. 9.

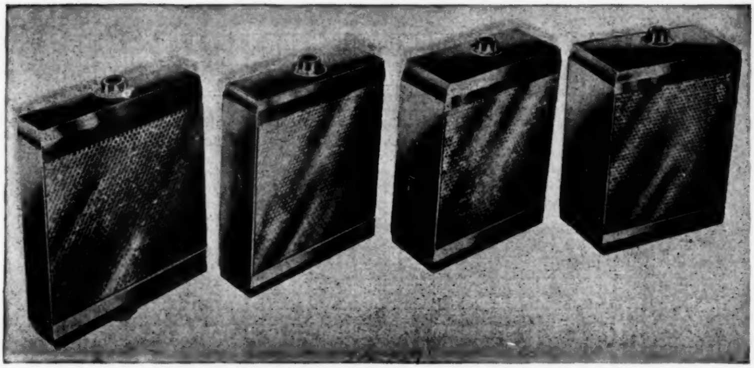


FIG. 5. HEATERS USED IN TESTS

The results given in Figs. 8 and 9 were computed by the use of the following formula:

$$H = \frac{KS\,(t_3 - t_1)}{1 + \dfrac{KS}{120\,AV_1Ds}} \qquad (5)$$

$$t_2 = t_1 + \frac{H}{60\,AV_1Ds} \qquad (6)$$

$$W_1 = \frac{H \times 960}{S} \qquad (7)$$

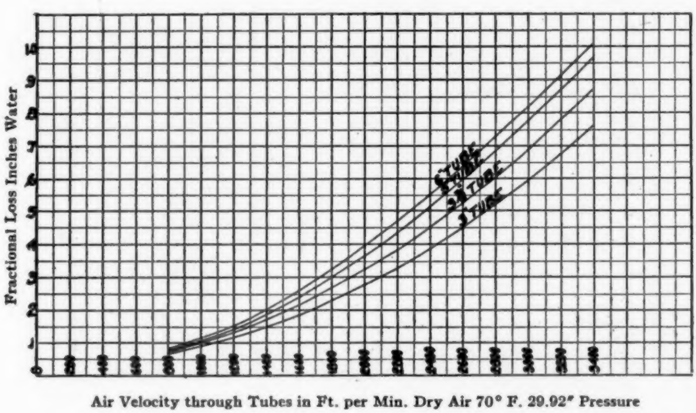


FIG. 6.

Heat Transfer Coefficient Velocity

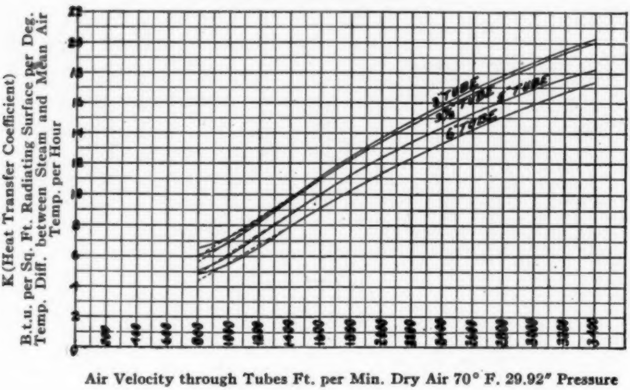


FIG. 7.

H = B.t.u.'s per hr.
t_1 = Temperature of air entering heater deg. fahr.
t_2 = Temperature of air leaving heater deg. fahr.
t_3 = 227.15 deg. fahr. steam temp. 5 lb. gage
A = Free area of heater (62.9 per cent)
V_1 = Velocity in ft. per min. through heater
D = Density of air lb. per cu. ft.
s = Specific heat of dry air = .2375
W_1 = Lb. condensate per hr. per sq. ft. surface.

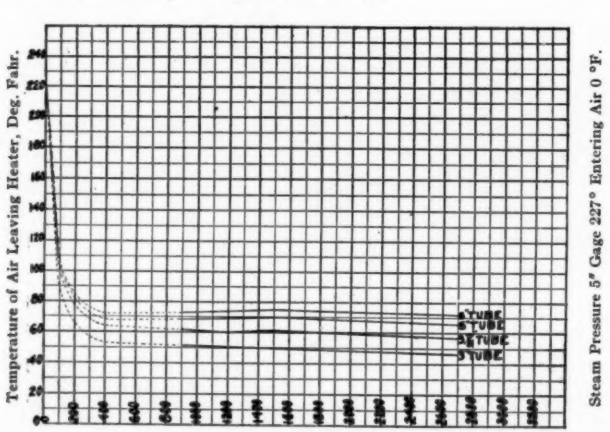


FIG. 8.

In Table 1 will be found complete data on the heaters used in these experiments.

TABLE 1. INFORMATION ON HEATERS USED

Designation	Length of tube inches	Dimensions core	Inches heater complete	Weight Lb.	Weight Oz.	Net surface in sq. ft.
54	3	12 × 12 × 3	16 × 12 × 3	23	4	27.12
53	$3^7/_8$	12 × 12 × $3^7/_8$	16 × 12 × $3^7/_8$	25	2	34.73
80	5	12 × 12 × 5	16 × 12 × 5	32	4	44.50
55	6	12 × 12 × 6	16 × 12 × 6	36	2	53.20

TABLE 2.

Date—11-25-24
Core 55 —0.6 in. tube
Run No. 12
Density of air
Actual 0.07197 lb. per cu. ft.
Dry 0.07215

Barometric pressure 29.85 in.
Relative humidity 12 per cent
Humid heat 0.2393

I. AIR MEASUREMENTS

Time min.	Humidity deg. fahr.	Vel. head 1–10	Static head 1–10	F. L. 1–5	T. in deg. fahr.	T. out deg. fahr.	T. rise deg. fahr.
0	91 dry	1.87	2.93	3.33	90.0	140.0	
2	60 wet				87.6	139.1	
4	31 diff.				89.0	139.5	
6		1.88	2.94	3.31	88.9	139.6	
9	12%				89.0	139.7	
11					89.1	140.0	
13					89.4	139.6	
16		1.88	2.92	3.32	88.8	139.2	
18					88.0	138.8	
20					88.0	138.7	
Av.		1.88	2.93	3.32	88.78	139.42	50.64
Av. Gasoline		0.185	0.273	0.634			
Av. Water		0.1367	0.2018	0.4686			
Av. Mercury			0.015				

II. THERMAL MEASUREMENTS

Time	Steam pressure 5 lb. gage	Steam T. in deg. cent.	Steam T. out deg. cent.	Condensate lb.
0	5	108.78	104.6	
		108.60		
		108.65		
		108.72		
7′ 36.6″	5	109.00	104.6	10
		109.08		
		109.20		
15′ 16.0″	5	109.40	104.8	20
		109.53		
		109.21		
		109.02		
22′ 51.6″	5	109.00	105.0	30
Av. 7.622′		109.02 deg. cent.	104.75 deg. cent.	10
		228.23 deg. fahr.		
	Correction	.56		
		228.79		
		227.15		
		1.64 deg. fahr. superheat		
		=0.76 B.t.u.		
		960.00		
		960.76 B.t.u. per lb. steam		

III. CALCULATIONS

(a) From Air Measurements

$$V = 1040\sqrt{\frac{0.1367}{0.07197}} = 1433 \text{ ft. per min.}$$

B.t.u. = $1433 \times 1 \times 0.07197 \times 0.2393 \times 50.64 = 1250$ B.t.u. per min.

(b) From Thermal Measurements

$$V = \frac{\text{B.t.u.}}{D \times s \times 1 \times (t_2 - t_1)} = \frac{1261}{0.07197 \times 1 \times 0.2393 \times 50.64} = 1445 \text{ ft. per min.}$$

$$\text{Cond. per min.} = \frac{10}{7.622} = 1.312 \text{ lb. per min.}$$

$1.312 \times 960.8 = 1261$ B.t.u. per min.

(c) 1445

$\frac{1433}{12}$ ft. per min. difference $\quad \frac{100 \times 12}{1445} = 0.83$ per cent variation in velocity

(d) 1445 ft. per min. = 1417 ft. per min. for dry air 70 deg. fahr. 29.92 in. pressure

Discussion

In discussing the data presented, it will be interesting to compare the characteristics of the air-tube type of heater with those of standard cast iron sections.

Professor Allen in a paper before this Society[1] presented data which showed that with cast iron heaters frictional loss varies as the square of the velocity. In

TABLE 3. COMPARATIVE DATA

Entering Air 0 °F 227 ° F. Steam

	Velocity ft. per minute through heater	700		900		1200	
1.	Type of heater	Cast iron 2 stacks $4\frac{5}{8}$" C 60" 18 sect.	Copper 6" Tube	C. I. 2 stacks $5\frac{3}{8}$" C 60" 10 sect.	Copper $3\frac{7}{8}$"	C. I. 2 stacks 5 " C 60" 9 sect.	Copper 6"
2.	Dimensions—gross	83" × 20" × 60"	84.37" × 6 × 40"	54" × 20" × 60"	66.1" × $3\frac{7}{8}$" × 40"	45" × 20" × 60"	51.5" × 6" × 40"
3.	Space occupied—cu. ft.	57.7	11.7	37.6	5.92	31.2	7.15
4.	Weight—lb.	4720	593	2624	299	2360	350
5.	Free area—sq. ft.	14.05		10.85		8.29	
6.	Volume air—cu. ft. per min.	9,830		9,760		9,950	
7.	Frictional loss	0.0590"	0.0593"	0.076"	0.078"	0.157"	0.157"
8.	Final air temperature deg. fahr.	78.5	83.8	55.0	69.4	58.0	83.22
9.	Cond. per hr. per sq. ft.	1.49	0.772	1.84	1.26	2.23	1.31
10.	Surface—sq. ft.	576	1186	320	598	288	700
11.	B.t.u. per min.	13,736	14,663	9,555	12,057	10,272	14,739
12.	Frontal area—sq. ft. without tanks	34.6	22.3	22.5	17.25	18.8	13.17
13.	Frontal area—sq. ft. complete		23.4		18.34		14.3

[1] Vol. 23 No. 3, JOURNAL A.S.H.&V.E. page 299.

the case of copper heaters under discussion, frictional loss varies as the 1.825th power of the velocity. In other words, the frictional losses are lower with copper neaters than with cast iron heaters. It is generally considered that a true comparison of different heaters should be made on the basis of equal frictional losses. With this in mind, we have plotted condensation per square foot of surface against frictional loss for both types of heaters. The results are shown in Figs. 11 and 12. It can be seen that the condensation rate for cast iron heaters is somewhat greater than that of copper heaters per square foot of radiating surface.

In the design of a heating system the problem is essentially to deliver a given number of cubic feet of air per hour to which have been added a specific number of B.t.u.'s. To do this economically the power consumed to circulate this air should be carefully considered; in other words, the frictional loss should be as small as possible. The following comparison of standard cast iron heaters with air-tube cellular type of copper heater is made on this basis. Complete data are given for each type of heater that will deliver a definite number of cubic feet of air per hour at equal velocities and equal frictional losses.

Considering cast iron heaters as 100 per cent, a further comparison is given in Table 4 on a percentage basis.

Summary

1. Complete data for the use of air-tube cellular type of copper heater are presented.

2. A comparison is made with standard cast iron heaters as regards size, weight, and performance.

TABLE 3. (CONCLUDED)

1500		1800		2000		2500	
C. I.	Copper	C. I.	Copper	C. I.	Copper	C. I.	Copper
2 stacks	5"	5 stacks	5"	2 stacks	5"	2 stacks	6"
5³/₈" C		5³/₈" C		5³/₈" C		5³/₈" C	
60"		60"		60"		60"	
12 sect.		10 sect.		9 sect.		7 sect.	
65" × 20" × 60"	78.4" × 5" × 40"	54" × 20" × 60"	66" × 5" × 40"	48" × 20" × 60"	59.7" × 5" × 40"	38" × 20" × 60"	47.9" × 6" × 40"
45.2	9.08	37.6	7.63	33.3	6.91	26.4	6.65
3150	460	2624	384	2360	345	1840	324
13.00		10.85		9.77		7.62	
19,500		19,510		19,510		19,010	
0.210"	0.212"	0.303"	0.300"	0.370"	0.36"	0.58"	0.59"
44.0	78.09	41.0	78.34	39.0	77.4	34.0	81.6
2.49	1.84	2.78	2.20	2.94	2.43	3.24	2.67
384	921	320	767	288	691	224	648
15,272	27,114	14,238	27,166	13,548	26,866	11,612	27,717
27.0	20.7	22.5	17.25	20.0	15.52	15.82	12.19
	21.8		18.33		16.6		13.3

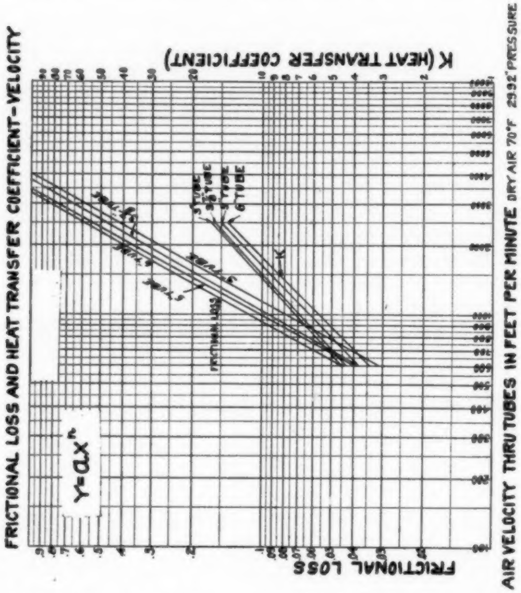

FRICTIONAL LOSS AND HEAT TRANSFER COEFFICIENT-VELOCITY
K (HEAT TRANSFER COEFFICIENT)
FRICTIONAL LOSS
Y=ax^n
FRICTIONAL LOSS
K
AIR VELOCITY THRU TUBES IN FEET PER MINUTE DRY AIR 70°F 29.92" PRESSURE

FIG. 10.

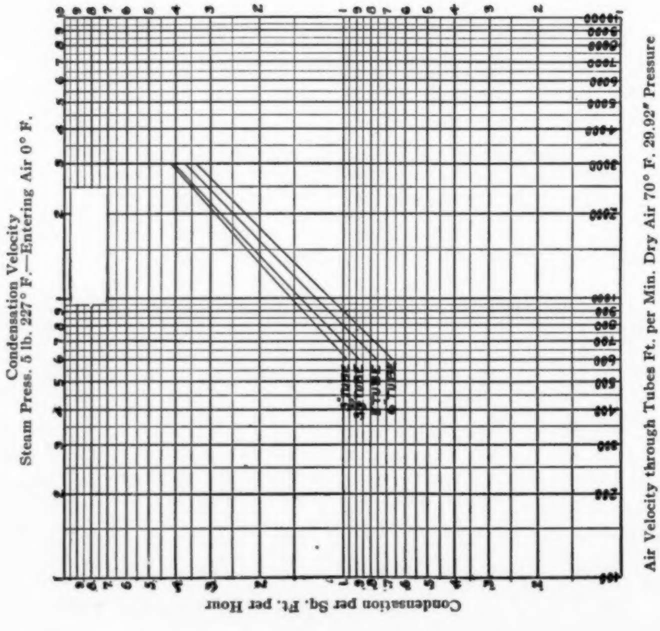

Condensation Velocity
Steam Press. 5 lb. 227° F.—Entering Air 0° F.
Condensation per Sq. Ft. per Hour
Air Velocity through Tubes Ft. per Min. Dry Air 70° F. 29.92" Pressure

FIG. 9.

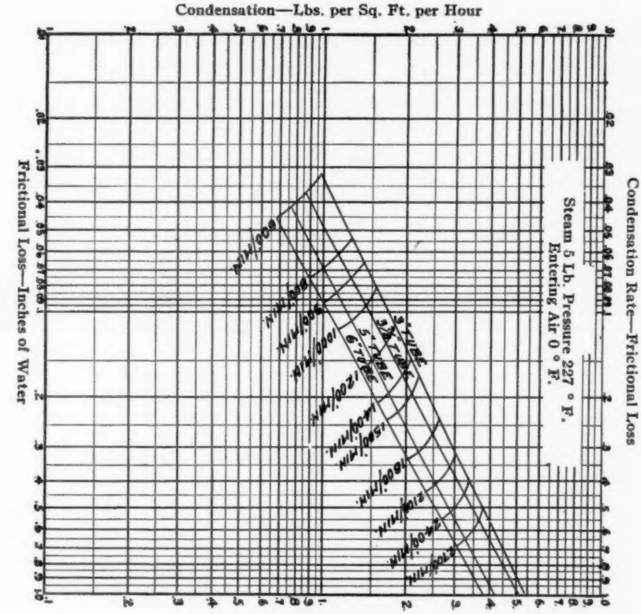

Condensation—Lbs. per Sq. Ft. per Hour
Condensation Rate—Frictional Loss
Steam 5 Lb. Pressure 227 ° F.
Entering Air 0 ° F.
Frictional Loss—Inches of Water

FIG. 11.

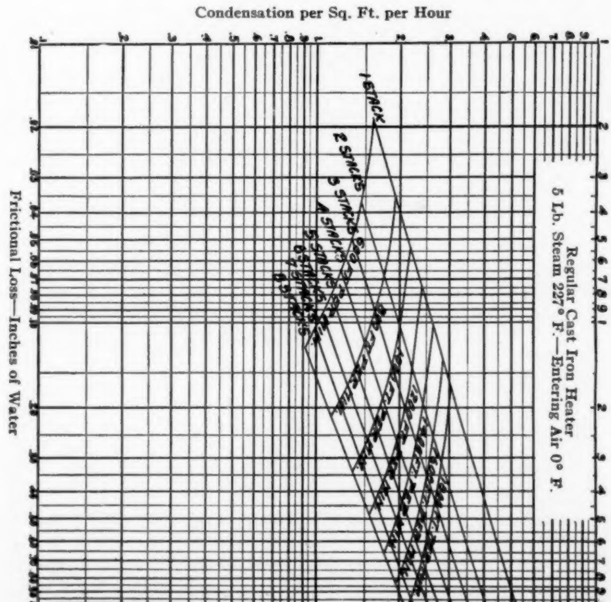

Condensation per Sq. Ft. per Hour
Regular Cast Iron Heater
5 Lb. Steam 227° F.—Entering Air 0° F.
1 STACK
2 STACKS
3 STACKS
4 STACKS
5 STACKS
6 STACKS
Frictional Loss—Inches of Water

FIG. 12.

TABLE 4. COMPARATIVE DATA

CONSIDERING THE CAST IRON HEATERS AS 100% EQUAL FRICTIONAL LOSSES

Vel. Ft. per min. through heater	700	900	1200	1500	1800	2000	2500
Per cent							
% Space occupied	20.4	15.7	22.9	20.1	20.3	20.8	25.2
% Weight	12.6	10.5	10.4	10.6	14.6	14.6	17.6
% Frontal area	71.3	84.8	77.2	84.5	84.5	85.5	84.7
% Net surface	206	187	243	240	240	240	289
% Final temp.	106.7	126.2	143.5	177.5	191.0	198.5	240.0
% Cond./hr./sq. ft.	51.7	68.4	58.7	73.9	79.1	82.7	82.5
% B.t.u. dissipated/min.	106.7	126.2	143.5	177.5	191.0	198.5	240.0

Conclusion

The authors wish to acknowledge the assistance of Professor E. H. Lockwood[1] for valuable suggestions and for his thorough inspection of the apparatus, methods of testing, and methods of calculation.

APPENDIX I

CALIBRATION DATA

The object of the exploration of the wind tunnel was to obtain the proper coefficient to use with the formula:

$$V = 1096\sqrt{\frac{p}{D}} \qquad (8)$$

V = Velocity in ft. per min.
p = Velocity pressure in inches of water
D = Density of air in lb. per cu. ft.

Velocity pressure readings were taken at 289 different positions for a given velocity. Fig. 13 is a diagrammatic sketch, showing the positions used in one section of the tunnel.

Figure 14 shows the results obtained for one velocity.

The method used in calculating the results was as follows: The square root of the velocity head readings were averaged and the square of this average taken in calculating the velocity in formula (8). This gave the average velocity of the air through the tunnel.

The comparison of the A. B. C. pitot tube with the pitot grid gave the coefficient (c) 0.949. Introducing this in the formula we obtained:

$$V = 1040\sqrt{\frac{p}{D}} \qquad (9)$$

The velocity determinations obtained by thermal measurements were calculated with the following formula:

$$V_1 = \frac{\text{B.t.u.}}{A \times D \times s \times (t_2 - t_1)} \qquad (10)$$

V_1 = Velocity in ft. per min. through heater
B.t.u. = B.t.u. per min.
A = Free area of core (62.9 per cent)
D = Density of air lb. per cu. ft.
s = Specific heat of dry air 0.2375
t_1 = Temperature of air entering heater
t_2 = Temperature of air leaving heater.

The coefficient (c_1) obtained by the thermal measurements was 0.9415. This value differs from (c) found by the air measurements by only 0.79 per cent.

[1] Associate Professor of Mechanical Engineering, Mason Laboratory, Yale University.

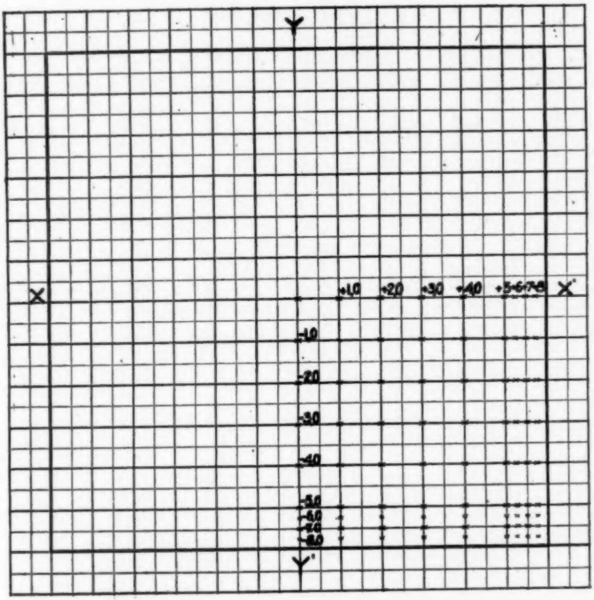
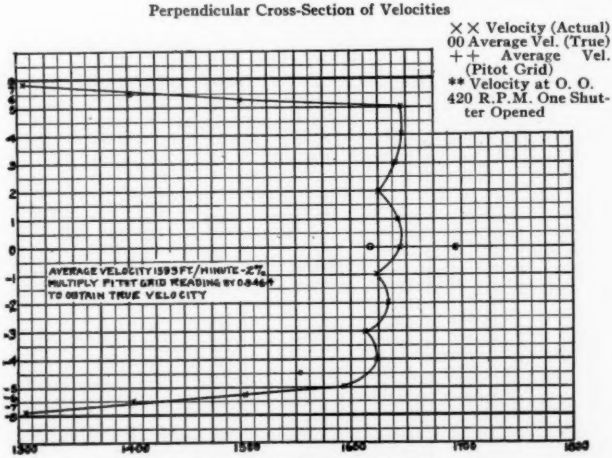


FIG. 13. SKETCH SHOWING POSITIONS AT WHICH READINGS OF VELOCITY HEAD WERE TAKEN

Perpendicular Cross-Section of Velocities



Velocity Ft. per Min. Dry Air 70° F. 29.92" Pressure

FIG. 14.

APPENDIX II

RESULTS OF TESTS

Heater No. 54—3 in. Tube—Net Radiating Surface 27.12 Sq. Ft.—Steam Pressure 5 Lb.

Run No.	*Vel. ft. per min.	F. L. in water	Cond. per min.	K	Temperature deg. fahr. Inlet	Outlet	Rise	Steam temp. Inlet ° fahr.
1	955	0.0813	0.584	7.032	25.82	75.67	49.85	227.86
2	1097	0.104	0.648	7.830	27.13	75.34	48.21	227.30
3	1340	0.138	0.727	9.31	38.57	83.64	45.07	228.09
4	949	0.0798	0.537	6.95	39.59	86.69	47.10	227.87
5	1432	0.157	0.780	9.96	38.14	83.27	45.13	227.84
6	1677	0.210	0.920	11.67	37.02	82.35	45.33	227.84
7	2370	0.407	1.253	15.74	36.00	80.00	44.00	228.04
8	3230	0.695	1.580	19.61	35.64	76.00	40.36	228.87
9	940	0.077	0.458	6.84	64.21	105.49	41.28	227.79
10	1170	0.108	0.537	8.02	65.29	104.56	39.07	227.55
11	1346	0.133	0.620	9.11	63.01	102.06	39.05	227.68
12	1509	0.163	0.686	10.22	65.14	103.70	38.56	227.71
13	1789	0.222	0.815	11.99	63.24	101.88	38.64	227.71
14	2129	0.3104	0.936	13.91	65.34	103.10	37.76	227.75
15	2685	0.498	1.171	17.10	63.11	100.11	37.00	228.89
16	3346	0.740	1.401	20.17	61.78	97.24	35.46	228.18
17	901	0.0769	0.391	6.68	83.97	121.53	37.56	227.73
18	1356	0.136	0.557	9.196	80.73	116.20	35.47	227.68
19	1692	0.200	0.684	11.39	82.04	116.97	34.94	227.68
20	2311	0.353	0.899	14.90	82.16	115.81	33.65	227.70
21	2961	0.503	1.04	17.16	82.73	115.03	32.30	227.73
22	3485	0.736	1.22	20.18	83.71	113.94	30.23	227.89
23	3411	0.727	1.263	20.46	80.11	111.93	31.82	227.72
24	2807	0.541	1.071	17.51	80.76	113.57	32.81	228.94

* Reduced to 70 deg. fahr. dry air 29.92 in. pressure.

Heater No. 53—3⁷/₈ in. Tube—Net Radiating Surface 34.73 Sq. Ft.—Steam Pressure 5 Lb.

Run No.	*Vel. ft. per min.	F. L. in water	Cond. per min.	K	Temperature deg. fahr. Inlet	Outlet	Rise	Steam temp. Inlet ° fahr.
1	3300	0.816	1.498	19.73	81.73	120.55	38.82	229.00
2	2445	0.483	1.445	15.81	87.05	127.00	39.95	228.70
3	1838	0.288	0.9386	12.40	80.23	124.09	43.86	230.40
4	1375	0.171	0.6808	9.64	88.73	131.26	42.53	232.30
5	908	0.0858	0.4454	6.35	89.50	131.76	42.26	233.40
6	881	0.0813	0.4331	6.18	89.44	132.31	42.87	227.38
7	856	0.07834	0.6128	6.393	39.48	96.03	56.55	227.68
8	1596	0.2173	1.085	11.21	38.26	94.69	56.43	228.20
9	2375	0.4435	1.518	15.42	37.26	90.30	53.04	228.56
10	2930	0.664	1.778	17.88	36.65	86.97	50.32	230.96
11	3290	0.806	1.936	19.22	36.86	85.93	49.07	236.10
12	1182	0.1360	0.5593	8.078	91.78	132.65	40.87	228.06
13	1615	0.2143	0.7787	11.21	91.01	132.88	41.87	228.04
14	1937	0.2956	0.9198	13.05	89.66	130.83	41.17	228.00
15	2235	0.395	1.043	14.63	88.51	129.00	40.40	228.72
16	2570	0.493	1.188	16.32	86.28	126.44	40.16	228.35
17	2773	0.576	1.266	17.36	85.35	125.95	39.60	228.43
18	3130	0.714	1.391	18.95	86.02	124.51	38.49	228.33
19	3238	0.768	1.432	19.37	85.33	123.53	38.20	229.08
20	2730	0.590	1.835	17.38	23.56	78.55	54.99	238.03

TABLE (CONCLUDED)

Run No.	*Vel. ft. per min.	F. L. in water	Cond. per min.	K	Temperature deg. fahr. Inlet	Outlet	Rise	Steam temp. Inlet ° fahr.
21	2222	0.4035	1.550	14.78	24.52	81.24	56.72	229.71
22	1616	0.2276	1.179	11.43	26.27	85.62	59.35	228.42
23	3265	0.798	2.052	19.54	26.59	78.08	51.49	234.50
24	2341	0.4435	1.408	15.52	51.40	101.75	50.35	228.58

* Reduced to 70 deg. fahr. dry air 29.92 in. pressure.

Heater No. 80—5 in. Tube—Net Radiator Surface 44.50 Sq. Ft.—Steam Pressure 5 Lb.

Run No.	*Vel. ft. per min.	F. L. in water	Cond. per min.	K	Temperature deg. fahr. Inlet	Outlet	Rise	Steam temp Inlet ° fahr.
1	2448	0.584	1.687	15.59	57.96	114.84	56.88	236.84
2	1870	0.349	1.366	12.34	53.03	114.20	61.17	230.02
3	921	0.0917	0.525	5.71	83.51	132.88	49.18	227.36
4	1655	0.254	0.954	10.38	83.16	133.04	49.88	227.78
5	2430	0.512	1.331	14.32	82.83	130.27	47.44	230.47
6	876	0.0887	0.663	5.22	42.76	105.38	62.62	227.90
7	1489	0.2409	1.197	10.09	41.80	105.26	63.46	228.33
8	1894	0.3356	1.424	11.95	41.78	103.98	62.20	228.13
9	2290	0.4819	1.674	13.97	41.57	102.26	60.69	229.62
10	2848	0.716	1.975	16.54	41.70	99.03	57.33	230.36
11	3160	0.864	2.139	17.73	41.34	97.88	56.54	245.39
12	1281	0.170	0.714	7.77	83.87	132.37	48.50	227.79
13	2032	0.365	1.149	12.42	82.26	131.41	49.15	227.42
14	2293	0.479	1.290	13.75	81.31	130.06	48.75	227.55
15	2663	0.610	1.442	15.26	81.41	128.28	46.87	228.14
16	2938	0.774	1.527	16.22	82.20	127.43	45.23	234.45
17	3158	0.836	1.650	17.43	81.07	126.62	45.55	239.29
18	3190	0.836	1.672	17.58	80.34	126.09	45.75	239.61
19	3174	0.836	1.669	17.49	79.74	125.57	45.83	242.33
20	3079	0.798	1.607	17.10	82.85	127.83	44.98	228.04
21	2595	0.615	1.434	15.21	81.63	128.56	46.93	227.86
22	2030	0.368	1.20	12.91	76.36	127.20	50.84	228.27
23	1425	0.203	0.867	9.07	77.32	129.22	51.90	228.42
24	3270	0.92	2.084	18.44	54.12	107.4	53.28	229.21
25	2920	0.773	1.923	16.96	52.54	107.62	55.06	232.95
26	2405	0.540	1.579	14.53	58.78	114.02	55.24	229.11
27	1679	0.282	1.130	10.72	62.10	119.05	56.95	228.60
28	910	0.108	0.592	5.72	65.43	120.73	55.30	232.27
29	1406	0.266	0.8418	8.97	79.87	131.17	51.30	228.31
30	2140	0.429	1.240	13.12	79.93	129.57	49.64	228.44
31	2392	0.506	1.364	14.30	79.14	128.00	48.86	229.10
32	2785	0.68	1.544	16.12	79.33	126.85	47.52	228.87
33	3155	0.888	1.684	17.61	80.15	126.00	45.85	232.41
34	848	0.098	0.5078	5.44	80.62	131.88	51.26	228.38
35	1138	0.148	0.6522	6.95	80.96	130.14	49.18	228.17
36	1140	0.148	0.6442	7.00	82.12	131.26	49.14	251.04
37	1122	0.140	0.6469	6.87	80.68	129.91	49.23	228.06
38	873	0.093	0.518	5.45	81.66	132.62	50.96	227.82
39	1709	0.281	1.023	10.91	80.16	131.44	51.28	227.82
40	1983	0.358	1.161	12.31	79.84	130.00	50.16	227.97

TABLE (CONCLUDED)

Run No.	*Vel. ft. per min.	F. L. in water	Cond. per min.	K	Temperature deg. fahr.			Steam temp. Inlet ° fahr.
					Inlet	Outlet	Rise	
41	1047	0.126	0.603	6.42	80.90	130.14	49.24	227.57
42	1378	0.189	0.819	8.66	79.44	130.12	80.68	227.64
43	1798	0.303	1.075	11.37	79.26	130.22	50.96	227.64
44	2795	0.584	1.57	16.44	79.40	127.48	48.08	228.13
45	1247	0.157	0.714	7.58	80.68	129.60	48.92	227.50
46	1540	0.246	0.934	9.87	78.92	130.55	51.63	227.97
47	909.3	0.104	0.541	5.76	80.20	130.97	50.77	227.76
48	1570	0.246	0.949	10.04	79.08	130.6	51.52	227.98
49	3043	0.375	1.655	17.39	80.14	126.76	46.62	235.54
50	893	0.0961	0.6806	5.78	43.12	105.96	62.84	228.15
51	1095	0.0983	0.8072	6.87	44.54	105.42	60.88	228.06
52	1317	0.252	0.9915	8.50	45.00	107.20	62.20	228.09
53	1591	0.354	1.207	10.40	45.50	108.15	62.65	228.22
54	2293	0.535	1.813	13.96	26.22	91.48	65.26	229.69
55	2375	0.570	1.869	14.40	26.50	91.50	65.00	229.43
56	3221	0.923	2.307	17.93	27.10	86.66	59.56	238.43
57	885	0.1005	0.752	5.66	20.19	90.06	69.87	227.36
58	1133	0.1419	0.9434	7.14	21.96	90.24	68.28	227.19
59	1279	0.1922	1.128	8.67	23.66	93.33	69.67	228.43
60	1473	0.2557	1.290	10.00	25.05	94.40	69.35	227.63
61	1962	0.3873	1.609	12.45	25.87	93.47	67.60	228.00

* Reduced to 70 deg. fahr. dry air 29.92 in. pressure.

Heater No. 55—6 in. Tube—Net Radiating Surface 53.20 Sq. Ft.—Steam Pressure 5 Lb.

Run No.	*Vel. ft. per min.	F. L. in water	Cond. per min.	K	Temperature deg. fahr.			Steam temp. Inlet ° fahr
					Inlet	Outlet	Rise	
1	850	0.0813	0.7023	4.93	38.55	107.19	68.64	227.48
2	1122	0.1271	0.8720	6.01	37.81	102.34	64.53	227.97
3	1253	0.1641	0.9915	6.85	38.01	102.95	64.94	227.68
4	1468	0.2143	1.179	8.21	38.37	105.07	66.70	227.60
5	1687	0.2823	1.381	9.66	38.13	106.19	68.06	227.69
6	2027	0.4065	1.647	11.59	39.17	106.91	67.74	229.89
7	2319	0.5140	1.857	12.98	37.77	104.57	66.80	242.24
8	3010	0.8280	2.310	15.94	36.97	101.21	64.24	244.17
9	873	0.0783	0.514	5.028	90.69	142.03	51.34	227.81
10	1327	0.158	0.739	7.041	89.32	137.74	48.42	227.54
11	1713	0.2764	1.01	9.685	88.56	139.78	51.22	228.42
12	2250	0.4690	1.312	12.56	88.78	139.42	50.64	228.79
13	2835	0.72	1.607	15.28	88.15	137.42	49.27	235.92
14	3101	0.826	1.740	16.32	87.28	135.96	48.68	229.98
15	2651	0.630	1.515	14.40	88.39	137.98	49.59	231.76
16	871	0.09756	0.6618	5.066	54.0	117.3	63.30	227.64
17	1885	0.343	1.387	10.67	55.32	117.48	62.16	227.96
18	2460	0.590	1.771	13.61	54.75	116.08	61.33	241.16
19	3074	0.886	2.106	16.23	56.80	115.07	58.27	236.66
20	862	0.099	0.745	4.96	28.74	100.13	71.39	227.59
21	2140	0.473	1.82	12.12	29.06	99.40	70.34	228.33
22	2935	0.860	2.36	15.60	29.64	96.20	66.56	232.22

* Reduced to 70 deg. fahr. dry air 29.92 in. pressure.

APPENDIX III

Final Temperatures and Condensations

Steam 227° F. 5 lb. gage; Air entering —10° F. to 70° F.

3 in. tube—Velocity through heater in ft. per min. measured at 70° F.—29.92 in. mercury

Temperature of entering air deg fahr.	900 F. T.	900 C.	1200 F. T	1200 C.	1500 F. T.	1500 C.	1800 F. T.	1800 C.	2100 F. T.	2100 C.	2400 F. T.	2400 C.	2700 F. T.	2700 C.
−10	51.38	1.43	49.62	1.85	48.31	2.26	47.89	2.69	47.12	3.10	45.93	3.46	44.44	3.79
0	58.79	1.37	57.16	1.77	55.85	2.16	55.44	2.58	54.72	2.96	53.57	3.32	52.15	3.63
10	66.15	1.31	64.54	1.69	63.34	2.07	62.95	2.46	62.25	2.84	61.16	3.18	59.80	3.47
20	73.56	1.26	72.03	1.61	70.89	1.97	70.51	2.35	69.85	2.70	68.81	3.03	67.51	3.31
30	80.98	1.19	79.52	1.54	78.43	1.88	78.08	2.24	77.44	2.57	76.45	2.88	75.21	3.15
40	88.39	1.13	87.00	1.46	85.97	1.78	85.64	2.12	85.04	2.44	84.09	2.73	82.92	2.99
50	95.81	1.07	94.49	1.38	93.52	1.69	93.20	2.01	92.63	2.31	91.74	2.59	90.63	2.83
60	103.2	1.01	102.0	1.30	101.1	1.59	100.8	1.90	100.2	2.18	99.38	2.44	98.33	2.67
70	110.6	0.95	109.5	1.22	108.6	1.50	108.3	1.78	107.8	2.05	107.0	2.30	106.0	2.51
F. L.	0.065		0.110		0.165		0.230		0.300		0.385		0.480	

F. T., final temperature of air leaving heater deg. fahr.—C., lb. condensate per sq. ft. per hour—F. L., frictional loss in water.

Final Temperatures and Condensations

Steam 227° F. 5 lb. gage; Air entering —10° F. to 70° F.

3 ½ in. tube—Velocity through heater in ft. per minute measured at 70° F.—29.92 in. mercury

Temperature of entering air deg. fahr.	900 F. T.	900 C.	1200 F. T.	1200 C.	1500 F. T.	1500 C.	1800 F. T.	1800 C.	2100 F. T.	2100 C.	2400 F. T.	2400 C.	2700 F. T.	2700 C.
−10	62.46	1.31	61.92	1.74	61.59	2.16	60.69	2.56	59.62	2.95	58.21	3.30	56.74	3.63
0	69.40	1.26	68.89	1.67	68.58	2.07	67.71	2.46	66.68	2.82	65.33	3.16	63.93	3.48
10	76.35	1.20	75.85	1.59	75.56	1.98	74.73	2.35	73.75	2.70	72.46	3.02	71.11	3.33
20	83.29	1.15	82.82	1.52	82.54	1.89	81.75	2.24	80.81	2.57	79.58	2.88	78.30	3.17
30	90.24	1.09	89.79	1.45	89.52	1.80	88.77	2.13	87.87	2.45	86.76	2.74	85.48	3.02
40	97.18	1.04	96.76	1.38	96.50	1.71	95.79	2.02	94.94	2.33	93.83	2.60	92.67	2.87
50	104.1	0.98	103.7	1.31	103.5	1.62	102.7	1.92	102.0	2.20	101.0	2.46	99.86	2.71
60	111.1	0.93	110.7	1.23	110.5	1.53	109.8	1.81	109.1	2.08	108.1	2.33	107.0	2.56
70	118.0	0.87	117.6	1.15	117.4	1.44	116.8	1.70	116.0	1.95	115.2	2.19	114.2	2.41
F. L.	.078		.130		.194		.270		.355		.450		.560	

F. T., final temperature of air leaving heater deg. fahr.—C., lb. condensate per sq. ft. per hour—F. L., frictional loss in water.

APPENDIX III (CONCLUDED)

FINAL TEMPERATURES AND CONDENSATIONS

Steam 227° F. 5 lb. gage; Air entering —10° F. to 70° F.

5 in. tube—Velocity through heater in ft. per minute measured at 70° F.—29 92 in. mercury

Temperatere of entering air deg. fahr.	900		1200		1500		1800		2100		2400		2700	
	F. T.	C.	F. T.	C.	F. T.	C.	F. T.	C.	F. T.	C.	F. T.	C.	F. T.	C.
−10	70.15	1.14	71.14	1.53	71.53	1.92	71.04	2.30	70.13	2.65	68.61	2.97	66.74	3.26
0	76.78	1.09	77.71	1.47	78.09	1.84	78.34	2.20	76.76	2.54	75.30	2.84	73.50	3.12
10	83.40	1.04	84.30	1.40	84.66	1.76	84.22	2.10	83.38	2.42	81.99	2.72	80.28	2.99
20	90.01	0.99	90.9	1.34	91.22	1.68	90.79	2.00	90.00	2.31	88.68	2.59	87.14	2.85
30	96.64	0.94	97.45	1.27	97.78	1.60	97.38	1.91	96.62	2.20	95.36	2.47	93.80	2.71
40	103.3	0.90	104.0	1.21	104.3	1.52	104.0	1.81	103.2	2.09	102.0	2.34	100.6	2.57
50	109.9	0.85	110.6	1.14	110.9	1.44	110.5	1.71	109.9	1.98	108.7	2.22	107.3	2.44
60	116.5	0.80	117.2	1.08	117.5	1.36	117.1	1.62	116.5	1.87	115.4	2.09	114.1	2.30
70	123.1	0.75	123.8	1.01	124.0	1.28	123.7	1.52	123.1	1.75	122.1	1.97	120.9	2.16
F. L.	0.083		0.140		0.213		0.300		0.400		0.511		0.640	

F. T., final temperature of air leaving heater deg. fahr.—C., lb. condensate per sq. ft. per hour—F. L., frictional loss in water.

FINAL TEMPERATURES AND CONDENSATIONS

Steam 227° F. 5 lb. gage; Air entering —10° F. to 70° F.

6 in. tube—Velocity through heater in ft. per min. measured 70° F.—29.92 in. mercury

Temperature of entering air deg. fahr.	900		1200		1500		1800		2100		2400		2700	
	F. T.	C.	F. T.	C.	F. T.	C.	F. T.	C.	F. T.	C.	F. T.	C.	F. T.	C.
−10	75.09	1.01	76.88	1.37	77.32	1.72	77.18	2.06	76.54	2.39	75.96	2.70	74.09	2.99
0	81.51	0.97	83.22	1.31	83.64	1.65	83.50	1.98	82.89	2.29	81.83	2.58	80.55	2.86
10	87.93	0.92	89.56	1.26	89.97	1.58	89.82	1.89	89.24	2.19	88.71	2.47	87.01	2.74
20	94.33	0.88	95.90	1.20	96.28	1.51	96.15	1.80	95.59	2.09	95.08	2.36	93.46	2.61
30	100.7	0.84	102.2	1.14	102.6	1.43	102.5	1.72	101.9	1.99	101.5	2.24	99.91	2.48
40	107.2	0.82	108.6	1.08	108.9	1.36	108.8	1.63	108.3	1.89	107.8	2.13	106.4	2.36
50	113.6	0.75	114.9	1.03	115.2	1.29	115.1	1.54	114.6	1.79	114.2	2.02	112.8	2.23
60	120.0	0.71	121.2	0.97	121.6	1.21	121.4	1.46	121.0	1.69	120.6	1.90	119.3	2.11
70	126.4	0.67	127.6	0.91	127.9	1.14	127.7	1.37	127.3	1.58	126.5	1.79	125.8	1.98
F. L.	0.0935		0.157		0.236		0.328		0.430		0.550		0.680	

F. T., final temperature of air leaving heater deg. fahr.—C., lb. condensate per sq. ft. per hour—F. L., frictional loss in water.

DISCUSSION

E. H. LOCKWOOD (WRITTEN): The authors of this paper have presented data on a new type of air heater, which will be a valuable addition to the existing information on this subject. The advantages and disadvantages of the copper type of heater will doubtless be brought out in the discussion of this paper.

The writer wishes to comment on some features of the heat dissipation theory employed in the paper. Experimental tests were made on sample heaters of 1 sq. ft. area, of depths from 3 to 6 in., placed in a wind tunnel where the necessary observations could be made at any desired air speed. The report contains observational data for thirty or more runs for each of the heaters, at varying air speeds and inlet air temperatures. From the observational data the authors have computed for each depth of heater a complete table of heater performance for air velocity from 900 to 2700 ft. per min. and for inlet air temperatures from —10 to 70 deg. fahr.

The method of computing the tables is clearly stated by the authors' formulas V, VI, VII, by means of which the final temperature and condensation can be directly computed for the various air velocities and temperatures. This method is sometimes called the arithmetic method because it makes use of the arithmetic mean temperature difference between steam and air. The arithmetic method is approximate, and is used as a substitute for the exact and more complex logarithmic temperature difference.

For air heating problems such as those in this paper the two methods give results that are almost identical, with an advantage for the arithmetic method that it is far more convenient for computing. It may be of interest to give the formulas for the two methods that their differences may be clearly seen.

ARITHMETIC METHOD

The heat dissipated per hour can be written in three ways, first in terms of the steam condensed, second in terms of the air heated, third in terms of the heat transmitted by the surface, using the heat transfer coefficient. Using the notation of this paper the expressions for the heat quantities are,

$$H = 960\, W_1 S \qquad (1)$$

$$H = 60\, A V_1 D s \qquad (2)$$

$$H = KS\left[t_3 - \frac{t_1 + t_2}{2}\right] \qquad (3)$$

By algebraic transformation W_1 and t_2 can be eliminated from the expression for the heat and then can be computed separately by the following formulas as given in the paper,

$$H = \frac{KS(t_3 - t_1)}{1 + \dfrac{KS}{120 A V_1 D s}} \qquad (V)$$

$$t_2 = t_1 + \frac{H}{60 A V_1 D s} \qquad (VI)$$

$$W_1 = \frac{H}{960 S} \qquad (VII)$$

After determining values of K for a given heater, with a sufficient range of air velocities and temperatures, the performance of the heater can be computed for any other conditions as desired.

As a numerical example the authors sample log sheet may be used, where the arithmetic mean temperature difference is,

$$t_3 - \frac{(t_1 + t_2)}{2} = 227.15 - \frac{(88.78 + 139.42)}{2} = 113$$

The arithmetic value of K is $\dfrac{60 \times 1261}{53.2 \times 113} = 12.58$ B.t.u. per sq. ft. per deg. per hr.

The performance of the heater will now be computed for air velocity 2290 ft. per min. and inlet air at 0 deg. fahr. The data are $t_1 = 0$, $t_3 = 227.15$, $K = 12.58$, $S = 53.2$, $D = .072$, $s = 0.24$, $A = 0.629$, $V_1 = 2290$.

By formula V, $H = 124{,}200$ B.t.u. per hr.
By formula VI, $t_2 = 83.2$ deg. fahr.
By formula VII, $W_1 = 2.43$ lb. per sq. ft. per hr.

In making these computations it was assumed that the value of K at low temperatures was not appreciably different for same air velocity, also that the value of the density D has the value for 70 deg. fahr. irrespective of the inlet air temperature.

LOGARITHMIC METHOD

The three heat quantities may be expressed in a similar way to the arithmetic method, using the same notation:

$$H = 960\, W_1 S \quad \ldots\ldots (4)$$

$$H = 60\, A V_1 D s \quad \ldots\ldots (5)$$

$$H = KS\left[\frac{t_2 - t_1}{\log_e \dfrac{t_3 - t_1}{t_3 - t_2}}\right] \quad \ldots\ldots (6)$$

The log mean temp. difference is, $\dfrac{50.64}{\log_e \dfrac{227.1 - 139.4}{227.1 - 88.78}} = 111.2$. The log value of K is, $\dfrac{60 \times 1261}{53.2 \times 111.2} = 12.82$ B.t.u. per sq. ft. per deg. per hr.

The logarithmic values differ by about two per cent from the arithmetic values in this case. It must not be assumed, however that the final error is as large as this, for when the log value of K is used in the log formula, the resulting value of H becomes almost identical with that computed by the arithmetic formula.

Owing to the $\log_e$ term it has been found impossible to eliminate the W_1 and t_2 as was done in the arithmetic method, hence there are no companion formulas for V, VI, and VII. A solution for H can be made by the logarithmic formula (6) by a laborious trial method, assuming successive values of t_2 each closer to the true value, until H and t_2 are in agreement. Values of H obtained in this way are in practical agreement with those found by the arithmetic method as before stated.

The things that will be borne out by the discussion will be, I hope, whether this type of heater is a practical and durable heater, will it stand 25 or 30 lb. of steam

pressure, will it last for two or three years without leaks occurring in this very light looking structure. Those questions of durability will certainly be very important ones.

H. R. SPOFFORD: I would like to ask if the tables will show the relative efficiency of this type of heating surface as compared with the standard cast iron pipe.

L'R. G. BOUSQUET: The table will show that and it will be just what you saw on the screen.

No. 724

HEAT TRANSFERENCE AND COMBUSTION TESTS IN SMALL DOMESTIC BOILER[1]

Report of Series of Tests Made with Various Fuels to Determine Heat Absorption and Other Characteristics of Sectional Boiler Construction

By H. W. Brooks,[2] M. L. Orr,[3] W. M. Myler, Jr.[4] and C. A. Herbert[5]

Pittsburgh, Pa.

Introduction

IN AN effort to aid small heating boiler designers, users and others interested in the more efficient utilization of fuels, a program of cooperative research has been initiated between the U. S. Bureau of Mines and the Research Laboratory of the American Society of Heating and Ventilating Engineers at the Bureau's Pittsburgh Experiment Station with a view toward determining experimentally certain data on boilers typical of the class of those generally rated below 2000 sq. ft. of radiation.

That there is a vast field for improvement in the thermal efficiency and practical operation of such boilers is evidenced by a series of 648 unpublished tests on various boilers consisting of base, fire-pot, dome and two sections or equivalent, conducted under the direction of the Bureau on behalf of another Government Department some years ago in which it developed that with loads at from 50 to 100 per cent of rating and with semi-skilled attention, the average thermal efficiency proved to be 57.9 per cent while at medium and low ratings the efficiencies proved to be well below 50 per cent. These averages represent tests with 648 representative lignites, bituminous coals, anthracites and cokes gathered from all parts of the United States. Tests on bituminous coals and lignites with semi-skilled attention revealed average efficiencies roughly 15 per cent lower than those on anthracite and coke. It is, therefore, broadly the object of these tests, *First,* through the application of known principles of combustion and the determination of necessary supplemental operating and design data, to make possible in practical operation efficiencies more nearly approximating those which can be attained with anthracite and coke under laboratory conditions, and *Second,* to aid in raising the efficiency of small boiler

[1] Published by permission of the Director, Bureau of Mines.
[2] Fuel Engineer, U. S. Bureau of Mines.
[3] Assistant Mechanical Engineer, U. S. Bureau of Mines.
[4] Formerly Assistant Chemical Engineer, U. S. Bureau of Mines.
[5] Laboratory Assistant, A.S.H.&V.E.

Presented at the Annual Meeting of the American Society of Heating and Ventilating Engineers, January, 1925, Boston, Mass.

operation with high volatile bituminous coal to somewhere near that obtained when burning anthracite or coke.

Acknowledgments

Messrs. Charles Schramm and J. P. Stein of the Bureau of Mines assisted in conducting the tests. Acknowledgment is made to the chemical section of the Bureau of Mines and particularly H. M. Cooper under whose direction all analyses of coal

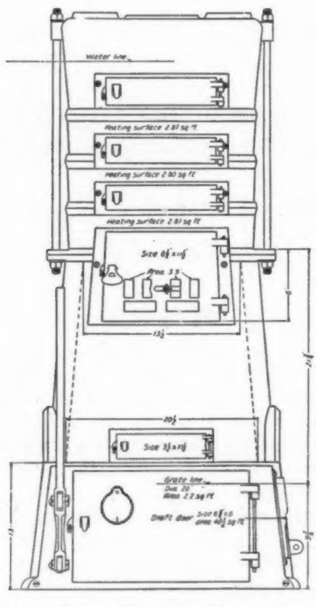


FIG. 1. FRONT VIEW OF COMPLETELY ASSEMBLED BOILER

were made. Acknowledgment is also due to Percy Nicholls, C. E. Augustine and R. A. Sherman of the Bureau of Mines for their aid and helpful suggestions in the preparation of the report.

Description of Previous Work

In the May, 1923, issue of the JOURNAL OF THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS was published a progress report of the results of this investigation as of that date on the typical, small round sectional boiler selected for test. It gave the results of the first two series of tests, the first series being those run with the base and fire-pot alone, and the second series being those run with the base, fire-pot and dome. The boiler selected was one having a 20-in. round grate of general dimensions shown in Fig. 1, equipped for tests as shown in Figs. 2 and 3. In each of these series the boiler was fired with coke, anthracite, bituminous coal and natural gas.

Specifically, the objects sought in the two previous series were (1) to determine the relative amounts of heat absorbed by the fire-pot and by the dome, (2) to determine the amount of free oxygen which passes from the ash-pit to the space above the fuel bed, and (3) to determine the value of ordinary methods of admitting air through the fire-door to burn combustible gases arising from the fuel bed.

An interesting series of curves was developed showing the relative thermal efficiencies and flue gas temperatures with the various fuels employed when using the fire-pot only as compared with the use of fire-pot and dome and when admitting secondary air through the slide provided in the firing door as compared with operation with slide in firing door closed. It developed that in the boiler under test,

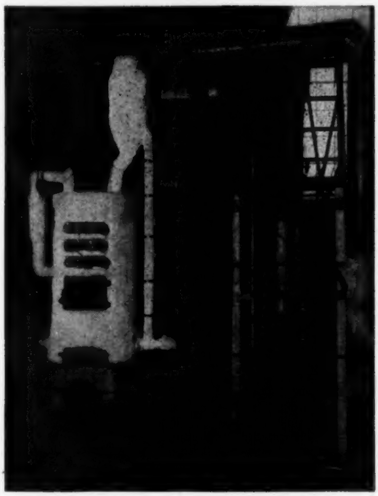


FIG. 2. COMPLETE BOILER ASSEMBLED AS USED FOR NATURAL GAS TESTS

free oxygen did find its way from the ashpit to the stack and consequently that with anthracite and coke the heat lost by not burning carbon monoxide was never over 5 per cent of the calorific value of the fuel. With bituminous coal, however, these losses were considerably higher though seldom throughout the first two series of tests was there insufficient oxygen to burn the combustible gases, vapors and soot, making it evident that inadequate provision was made for igniting, mixing and burning the combustibles arising from the fuel bed.

The tests showed that the admission of secondary air over the fuel bed to burn the carbon monoxide decreased the thermal efficiency when burning both anthracite and coke at all rates tested, since the small amount of carbon monoxide present did not produce sufficient heat in burning to carbon dioxide to compensate for that necessary to raise the temperature of the excess air content of the secondary air drawn in through the slots. However, with bituminous coal, admission of secondary air raised the efficiency at the high ratings and lowered it at the lower ratings. The tests further showed after firing and particularly after stirring the fuel bed

that secondary air can probably be admitted to advantage. They showed distinct limitations to the value of firing by the coking method as compared with firing by the spreading method in fire-pots of limited capacity.

Over-all thermal efficiencies, when burning anthracite and coke, varied between 51 and 59 per cent with the fire-pot alone, between 57 and 60 per cent when the dome was added to the fire-pot and ports in the fire-door kept closed, and between 49 and 58 per cent with the ports open. With bituminous coal the efficiency was much lower varying between 33 and 40 per cent with the fire-pot only, between 41

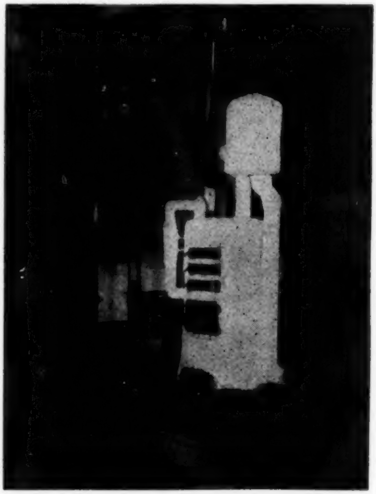


FIG. 3. COMPLETE BOILER ASSEMBLED AS USED FOR SOLID FUELS

and 48 per cent with the fire-pot and dome and no secondary air, and between 40 and 44 per cent with a supply of secondary air. On firing by the coking method efficiency varied between 38 and 43 per cent with no secondary air and between 41 and 46 per cent with secondary air.

When burning natural gas the efficiency varied between 40 and 57 per cent for the fire-pot only and between 55 and 63 per cent when the dome was added to the fire-pot.

Objectives of Present Series of Tests

During the past year a third series of tests has been run on the completely assembled boiler consisting of base, fire-pot, three intermediate sections and a steam dome. As before, the fuels used were coke, size $^3/_4$ to 3 in.; anthracite, stove size; Pittsburgh high volatile bituminous coal, size $^3/_4$ to 3 in.; and natural gas of 1105 B.t.u. per cu. ft. The purposes of the present series of tests were, *First*, to determine the additional amount of heat absorbed by the addition of three sections, *Second*, to verify and expand the data previously obtained regarding the passage of free oxygen from the ash-pit to the space above the fuel bed and, *Third*, to further as-

certain the value of ordinary methods of admitting secondary air over the fuel bed, as well as to secure further comparative data on the four fuels employed when using the complete boiler.

Heat Absorption of Sections

The heat absorption of the three sections added is most conveniently shown in the comparative thermal efficiencies of the present series compared with the two previous series.

The heat accounts of the coke tests for the complete boiler are shown graphically by Figs. 4 and 5, ports in firing door closed and open, respectively. The heat accounts for these, as well as for all following tests, have been based on the fuel ac-

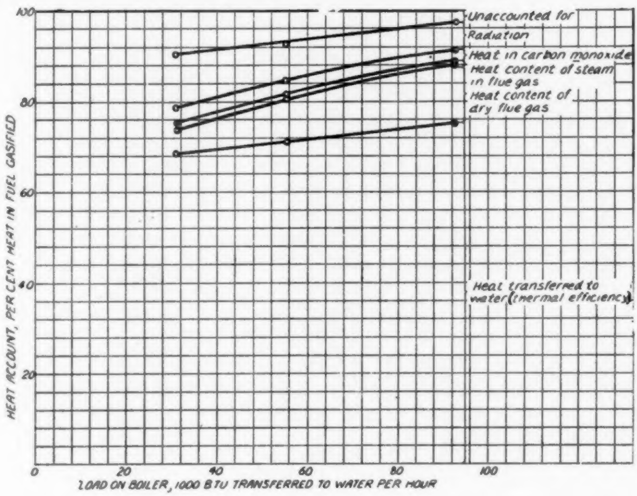


FIG. 4. HEAT ACCOUNT BASED ON HEAT IN FUEL GASIFIED, COKE. THIRD SERIES, COMPLETE BOILER. NO SECONDARY AIR, PORTS IN FIRE-DOOR CLOSED

tually gasified to avoid any confusion which might arise through introducing the losses caused by combustible being removed with the refuse from the ash-pit. Over-all thermal efficiencies, including ash-pit losses, are also supplied, for the benefit of those interested, in Item 51 of Table 2, but in the curves and the discussion the term thermal efficiency will refer to the per cent of heat generated in the grate actually transferred to the water, thus permitting comparisons based directly on the basis of heat actually transferred.

The distance from the base line to the first curve represents the thermal efficiency. The distances between subsequent curves, reading upward, represent, respectively, the dry gas loss, the loss to steam in the flue gas, the loss due to unburned carbon monoxide, and the radiation and convection losses. The distance between the upper curve and the top of the figure represent the losses unaccounted for.

The upper group of curves of Fig. 6 show the increase in thermal efficiency resulting from the addition of the 3 sections, which are, with closed slots, from 7 to 11 per cent over those of the fire-pot and dome and from 10 to 17.5 per cent over those obtained with the fire-pot only. With slots open the gain in the efficiency of the complete boiler over that with fire-pot and dome only ranges from 11 to 15 per cent.

The upper group of curves of Fig. 7 show the trend of increase of efficiencies with increase of heating surface which tend to indicate that even with three sections the most efficient boiler structure has not yet been attained for the particular ratings investigated when burning coke. These afford an interesting comparison with similar curves for anthracite (Fig. 11, which will be discussed later) where maximum efficiencies may be attained at lower ratings with a lesser number of sections added.

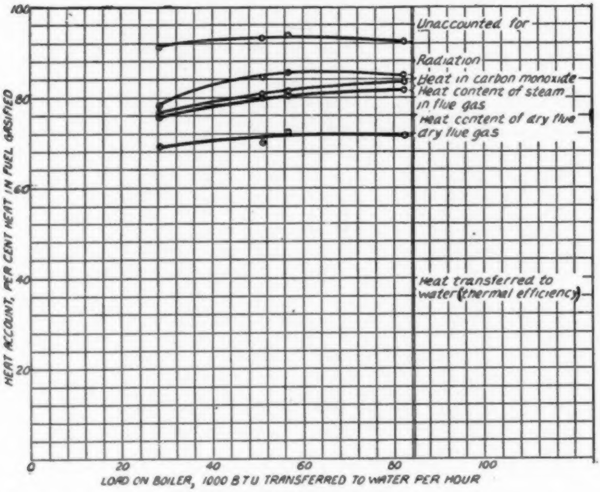


FIG. 5. HEAT ACCOUNT BASED ON HEAT IN FUEL GASIFIED, COKE. THIRD SERIES, COMPLETE BOILER. SECONDARY AIR PORTS IN FIRE-DOOR WIDE OPEN

The coke requires more sections for maximum efficiency than anthracite in this boiler and is undoubtedly due to the greater volume of the fire-pot occupied by the coke, thus leaving less volume above the fuel bed for completion of combustion of the partially burned gases.

The heat accounts for anthracite with slots in firing door closed are shown on Fig. 8, while the series with slots in firing door open are shown on Fig. 9.

As will be observed from the efficiency curves for anthracite shown at the top of Fig. 10, plotted from tests run with the slots in the firing door closed, the three additional sections increased the thermal efficiencies, based on heat in the fuel gasified, from 7 to 11 per cent over those of fire-pot and dome and from 11 to 17.5 per cent over those obtained with fire-pot only, saving increasing with the load. With the exception of tests 73 and 97, the thermal efficiencies on the present anthracite series (slots closed) varied from 68.3 to 73.5 per cent of the heat in the fuel gasified. Test 73 was run at a higher rate than any employed in this or the two previous

series with a view toward determining the action of the boiler at or near the manufacturers' rating. This test showed the low efficiency of 64.1 per cent, accounted for by the fact that the carbon monoxide losses were much greater as might be expected when operating at high rating. (Note also large increase in ash-pit losses at this high rating.) Consideration of the form of the typical boiler curve indicates that it is quite proper that efficiencies at maximum rating should be low in order to permit the range of maximum efficiencies at the lower loads carried during the major portion of the heating season. Average experience, extending over a period of years, shows that the heating season consists of approximately 100 days of moderate weather, 80 days of cold weather and 40 days of very cold or high windy weather. The latter period which constitutes only about 18 per cent of the entire heating season is the period at which a boiler will be called upon for its maximum rating. Obviously, efficiency may be sacrificed for this relatively short period if a corresponding gain is effected over the range of temperatures normally expected during the other 82 per cent of the season. When running with the slot open, the anthracite efficiencies, based on fuel gasified, with fire-pot and dome only, varied from 52.6 to 60.2 per cent while during the present series with the three sections added these figures increased to the range of 57.5 to 72.3 per cent, depending upon the load. Comparative curves, similar to those of Fig. 10, however, have not been repeated for operation with the introduction of secondary air over an anthracite fuel bed for, except at very high ratings, it does not represent the most effective practice as demonstrated by both the previous and present series of tests.

The trend of increase of efficiencies with the addition of heating surface and increased length of gas travel from the fuel bed to stack with the three loads of 28,000, 50,000 and 72,000 B.t.u. transferred to water per hour is indicated by three efficiency curves on Fig. 11. These curves are plotted from points from the thermal efficiency curve of Fig. 10 so that they may be on the basis of common loads.

The form of these curves tends to indicate that at the low and medium loads tested when burning anthracite in this particular boiler, with the slots closed, maximum efficiency would probably be reached with base, fire-pot, dome and probably not in excess of one section. It is a well-known fact among practical boiler men that at low rates of output with the ordinary boiler the top section can generally be removed and the efficiency raised but when this is done the efficiency at the higher ratings would be lower than before it was removed. This is amply demonstrated by the full rate tests numbers 73 and 97 in the latter of which, with slots in firing door open, the full advantage of the addition of sections becomes apparent. At low and medium ratings, such as were used on these tests, however, the decrease of efficiency by adding sections results from the increased loss of heat by radiation and convection which loss is greater than the gain in the heat imparted to the sections by the hot gases passing over them. In designing the most economical boiler the number of sections should thus be chosen so that the efficiency will be highest at the rating at which the greatest quantity of fuel is burned throughout the season.

The heat accounts of the bituminous tests for the complete boiler are shown on Fig. 12, with the ports in the firing door closed and on Fig. 13, with the ports in the firing door open. As in the two previous series of tests, the efficiencies are seen to be much lower than when firing anthracite or coke, principally because the unaccounted-for losses are much higher, and in a less degree on account of the greater loss due to the greater quantity of steam formed from the bituminous coal. The high unaccounted-for losses are largely due to the presence of gaseous hydrocarbons, hydrogen, tar and soot in the flue gases and as the order of these losses

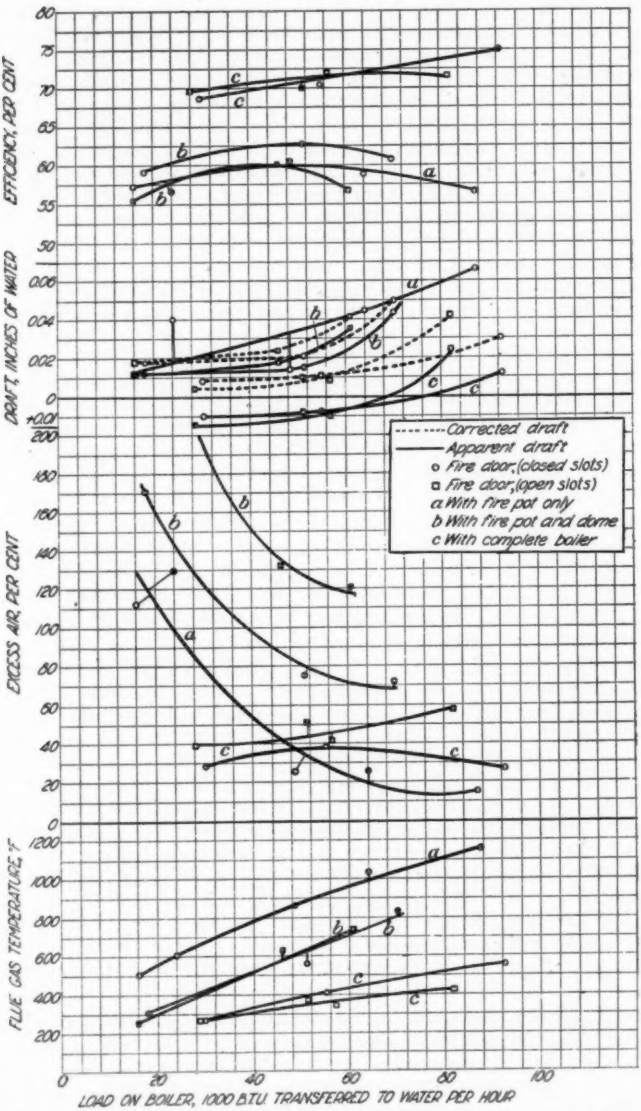


FIG. 6. CURVES SHOWING THE THERMAL EFFICIENCIES BASED ON FUEL GASIFIED, DRAFTS, THE TEMPERATURES AND EXCESS AIR CONTENTS OF THE FLUE GASES, FOR COKE

has been quite well established in the previous series and as they are apparently no greater in the present series, it was not considered worth while to repeat this special portion of the tests on the complete boiler.

The increase of efficiency on the present series of bituminous tests over the pre-

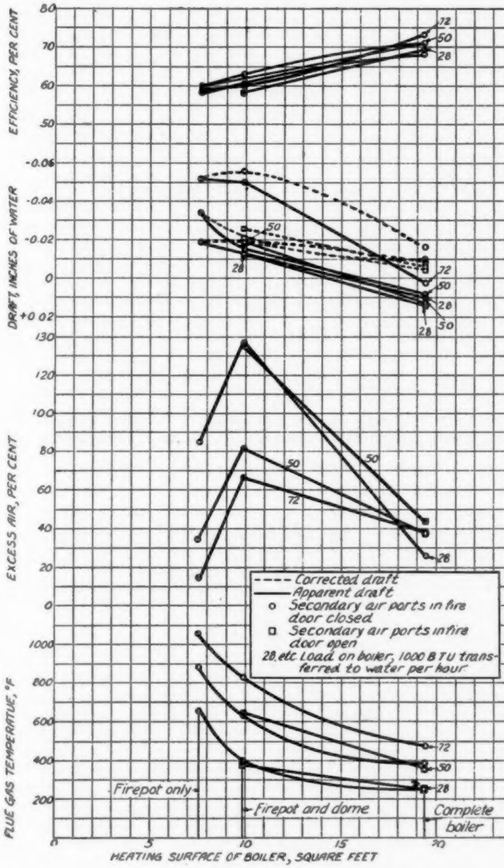


FIG. 7. CURVES SHOWING THE THERMAL EFFICIENCIES BASED ON FUEL GASIFIED, THE DRAFTS, TEMPERATURES AND EXCESS AIR CONTENTS OF THE FLUE GASES, WITH COKE, FOR THREE DIFFERENT LOADS

vious series is shown by the upper group of curves on Fig. 14. At low ratings complete boiler efficiencies are but slightly better than those with fire-pot and dome and from 9 to 15 per cent better than those with the fire-pot only. This again illustrates the fact that additional sections at low ratings do not serve to better operating

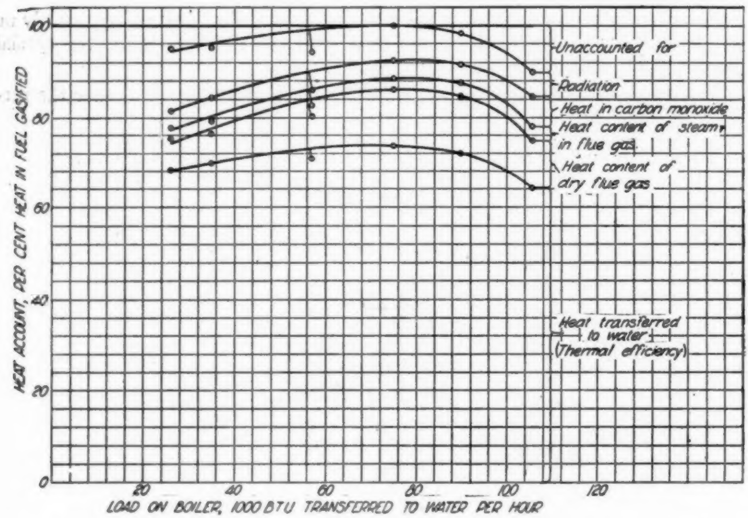


FIG. 8. HEAT ACCOUNT BASED ON FUEL GASIFIED. THIRD SERIES OF TRIALS, WITH ANTHRACITE, COMPLETE BOILER. SLOTS IN FIRING DOOR CLOSED

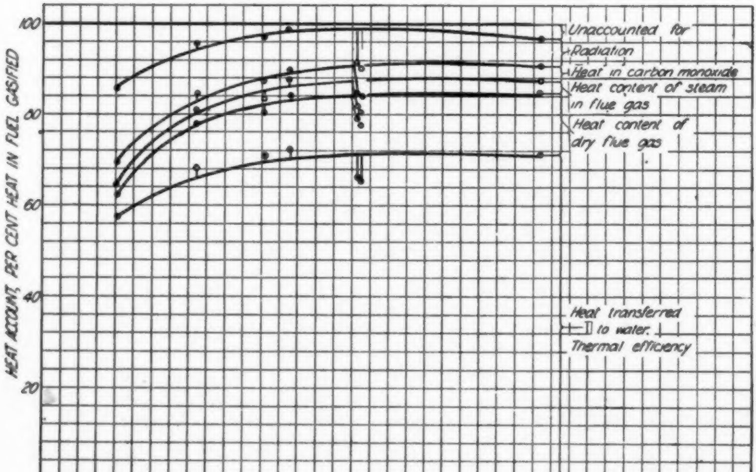


FIG. 9. HEAT ACCOUNT BASED ON FUEL GASIFIED. THIRD SERIES OF TRIALS, WITH ANTHRACITE, COMPLETE BOILER. SLOTS IN FIRING DOOR OPEN

efficiencies as the heat radiated and convected therefrom is about as great as and sometimes greater than their additional heat absorption. At the higher ratings, however, the complete boiler showed an increase in efficiency over the fire-pot and dome with slots closed of as high as 9 per cent over the tests with fire-pot and dome and as high as 10.5 per cent with fire-pot only. With the slots open the increase of efficiency of the complete boiler over that of the fire-pot and dome at the higher ratings is of the order of 7.5 per cent.

The trend of increase of these efficiencies with the addition of sections was shown in the upper set of curves of Fig. 15. As would be expected the slope of these curves is greater and the efficiencies continue rising with all three of the loads selected indicating that when using bituminous coal the point of maximum efficiency, even with the addition of three sections, has not yet been reached and that it is probable that four to five sections would be necessary to secure maximum efficiency when burning bituminous coal. Comparison of the slope of these curves with similar curves for anthracite of Fig. 11 stresses the necessity of increasing combustion volumes and length of gas travel by the addition of sections when converting the boiler previously fired with anthracite to the use of bituminous coal.

The detailed heat accounts for the natural gas trials have not been plotted, same having been carried out primarily to obtain values for the radiation and convection losses when the complete boiler was operated at a fairly high rating. The high unaccounted-for losses at the lowest rate of steaming are probably caused by an unsteady flame which permitted some gas to pass off unburned and undetected by analysis. In operating this boiler great care was required in adjusting the air supply in order to maintain the excess air at a value to insure fairly good efficiency. Had the boiler been operated otherwise the efficiency would have been very low and since with careful adjustment the over-all efficiency was only about that obtained with anthracite and coke, the necessity for having a carefully designed burner whereby the air supply be automatically and accurately adjusted is obvious if reasonable efficiency is to be maintained.

The efficiencies with natural gas during the previous series of tests varied from 55 to 63 per cent and during the present series from 45 to 75 per cent. With the exception of the low rate, where the efficiency dropped from 55 per cent in the previous series to 45 per cent in the present series, all of the efficiencies of tests at corresponding rates showed an increase of 10 per cent or better attributable to the addition of the three sections. Comparative curves showing these increases of efficiency with increased number of sections are shown on Fig. 16.

Passage of Free Oxygen Past the Fuel Bed

As in the previous series when burning coke and anthracite with the slot closed so as to shut off secondary air infiltration, free oxygen from the ash-pit was always present in the stack gases owing to passage of air through the outer edges of the fuel bed which air stratified and remained unmixed with the partly burned combustible gases above the fuel bed. The loss due to unburned carbon monoxide never exceeded 5 per cent (varying from 2.5 to 4.9 per cent with the load) with the exception of the special high rate anthracite test 73 where the high carbon monoxide loss of 6.8 per cent would normally be expected. Here the fire-pot was crowded with fuel, hence not permitting the continuance of high temperatures at the top of the fuel bed to aid in igniting the unburned carbon monoxide over the exposed incandescent fuel bed for such a long period of time as with the more normal charges. Again the available combustion volume and length of gas travel was insufficient to

permit thorough mixing, ignition and combustion of the large volume of partly burned gases released before they had been cooled below ignition temperatures in the flues. The decrease in carbon monoxide losses with increasing loads up to the most efficient rating of the boiler and subsequent increase thereafter when burning anthracite is shown in the carbon monoxide curves of Fig. 17. These curves show that up to the rating of maximum efficiency the higher the heat transfer rate, the higher the fuel bed temperature, the greater the turbulence of gas flow hence the better the mixing and the more complete the combustion of the gases as long as there is sufficient combustion volume and length of gas travel to permit. After the rating of maximum efficiency however, in spite of the higher temperatures within the fuel bed, the limited combustion space becomes so crowded with partially burned gases as to make thorough mixing impossible before the relatively cool heating surface is reached when gas temperatures fall below those necessary for further ignition thus quenching subsequent combustion. The consequent increase in carbon monoxide losses is one of the principal contributing factors to the droop in the boiler and furnace efficiency curve after the rating of maximum efficiency has been passed. The carbon monoxide curves for the coke tests on Fig. 16 show drooping characteristics, apparently not having been extended to the rating of maximum efficiency as is indicated from the coke test efficiency curves at the top of the sheet.

The fact that carbon monoxide was present despite the presence of sufficient oxygen in the flue gases to burn the carbon monoxide to carbon dioxide, coupled with proofs[6] of previous tests conducted by the Bureau that the fuel bed of a furnace acts as a gas producer and that the oxygen is all used up as it passes through the first 3 to 4 in. of the fuel bed, clearly indicates that the excess of oxygen present must have partly leaked in around the firing door and partly passed up near the outer water cooled boundary of the fuel bed and remained stratified around the outer walls of the fire-pot until forced to mix with the unburned combustible gases in the flues at which point the temperatures attained so decreased through radiant and convected heat to the heating surfaces that subsequent temperatures were not sufficiently high to ignite such combustible gases, thus causing them to pass out as diluted, relatively low temperature stack gases.

In the present series the temperatures of the products of combustion leaving the fire-pot were as low as 200 to 300 deg. fahr. at the lowest rates of combustion and rose only to 500 to 600 deg. fahr. at the highest rates of combustion. All of these temperatures are far below either the ignition temperature of carbon monoxide (about 1220 deg. fahr.), of methane (about 1200 to 1380 deg. fahr.), or of hydrogen (about 1080 deg. fahr.). Judging from the low stack temperatures it is probable that little combustion continued after the gases had passed through the first and possibly the second section. By far the major portion of the heat release must be affected within the fire-pot itself or else it will never be affected.

The problem of complete combustion, therefore, except at very high ratings, in such a furnace burning coke or anthracite at low or moderate ratings, consists not in adding further cold secondary air through the slots in the firing door which only serves to further dilute and cool the stack gases but rather to properly mix the free oxygen filtering around the sides of the fuel bed with the unburned combustible gases at a point near the surface of the fuel bed where temperatures are sufficiently high to permit their ignition. It should also be noted in connection with the introduction of secondary air, even in the case of anthracite or coke at high ratings or

[6] Kreisinger Henry, *et al.*, "Combustion in the Fuel Beds of Hand Fired Furnaces." Bureau of Mines Technical Paper 137.

high volatile bituminous coal where it is actually needed, that same possibly may enter through the open slots at such a low velocity that it would never have a

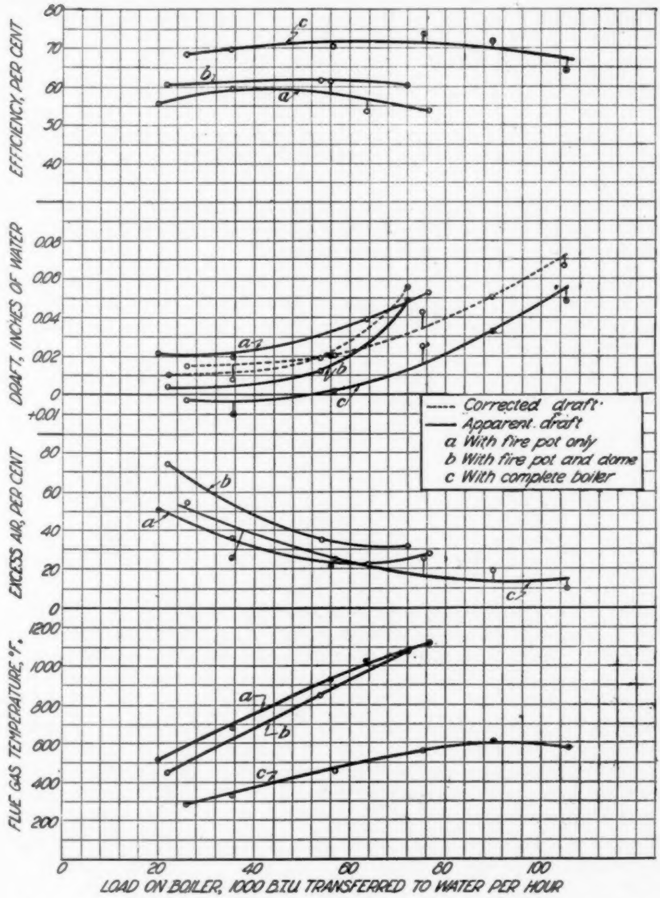


FIG. 10. CURVES SHOWING THE THERMAL EFFICIENCIES BASED ON FUEL GASIFIED, DRAFT, TEMPERATURES AND EXCESS AIR CONTENT OF FLUE GASES FOR ANTHRACITE. SECONDARY AIR PORTS IN FIRE-DOOR CLOSED

chance to combine with the greater portion of the carbon monoxide at the rear of the fuel bed.

The decreasing air infiltration around the walls of the fire-pot with the slots closed is reflected in the excess air curves of Figs. 10 and 11 which run fairly consistently as to shape. The greater the heat absorption, the faster the combustion, the greater the turbulence and mixing of gases, and the less the chance for formation of cool

non-burning area of low air resistance immediately adjacent to the fire-pot periphery, or in other words, the tighter the air seal between ash-pit and furnace. Comparative excess air curves of Fig. 11 which are plotted at equal loads of 28,000, 50,000 and 72,000 B.t.u. for the three series of tests indicate a trend of air infiltration more or less under the control of the furnace erector and the operator rather than phenomena entirely resultant from combustion conditions. However, it is noted that with flue gas temperatures roughly the same with the series 1 (fire-pot only) and series 2 (the fire-pot and dome) the excess air increases with increase of heating surface and consequently greater opportunity for over-fire leakage. As flue gas temperatures decrease, however, drafts decrease and correspondingly the forces tending to pull excess air both around and in over the fuel bed. This, coupled with probably a tighter furnace set-up in the present series than in the preceding series serves to explain the subsequent droop of these curves after the maximum point with the addition of heating surface.

In order to ascertain the effect on the carbon monoxide losses when using a low density fuel such as coke, a special series of tests which are not reported in detail herein were run using a small size coke. These tests showed a surprisingly high carbon monoxide loss.

The size of the coke used on the special tests as compared with the size used on the coke tests reported herein is given as follows:

Size of Screen, Inches	Small Coke	Large Coke
Through 3	...	100%
Through 2¼	...	90%
Through 2	...	62%
Through 1½	100%	22%
Through 1	62%	2%
Through ¾	Not used	Not used

The carbon monoxide loss using the large coke varied from 1.7 to 3.7 per cent of the heat in the fuel gasified, the average being 2.8 per cent in the test results reported herein. With the small size coke on the special tests the carbon monoxide losses varied from 7.9 per cent to 21.6 per cent, the average being 14.2 per cent. Similar results of increase in carbon monoxide content with decrease in the fuel size are reported in other similar tests in the June 23, 1923, issue of *Iron Age*, Fig. 6, page 1843, a complete discussion being given on page 1842. In order, therefore, to secure efficient combustion in any furnace one must be sure that the percentage of fine sizes is not so great as to completely seal off the ash pit from the space above the fuel bed for when this is done the producer effect set up within the furnace makes higher efficiencies well nigh impossible. Even when secondary air was introduced through the opening of the slots in the firing door, though there was a considerable improvement, efficiencies still did not approach the efficiencies with the proper size coke, for the lower fuel bed temperatures, the lesser exposure of the incandescent fuel bed to the partially burned gases above it, and the lesser turbulence of flow with the denser fuel bed, all reacted against proper operating conditions.

By comparing the curves on Fig. 16 of carbon monoxide loss and excess air for bituminous coal, here again it will be seen that at all times there was uncombined oxygen in the flue gases although unburned carbon monoxide was also present at the same time. As in the case of the anthracite and coke tests this is obviously due to incomplete mixing of the two, so that combustion was not completed before they came in contact with the cooler surface of the boiler. The tests with the slots in the firing door closed showed from 0.5 to 3 per cent greater loss than those with the slots

open, indicating that with the higher volatile content necessary to be distilled and burned with the bituminous coal, together with the tendency of the air leaking from the ash pit to the combustion chamber to remain stratified near the walls, that it is absolutely necessary, even at the lowest bituminous rates tested, to introduce

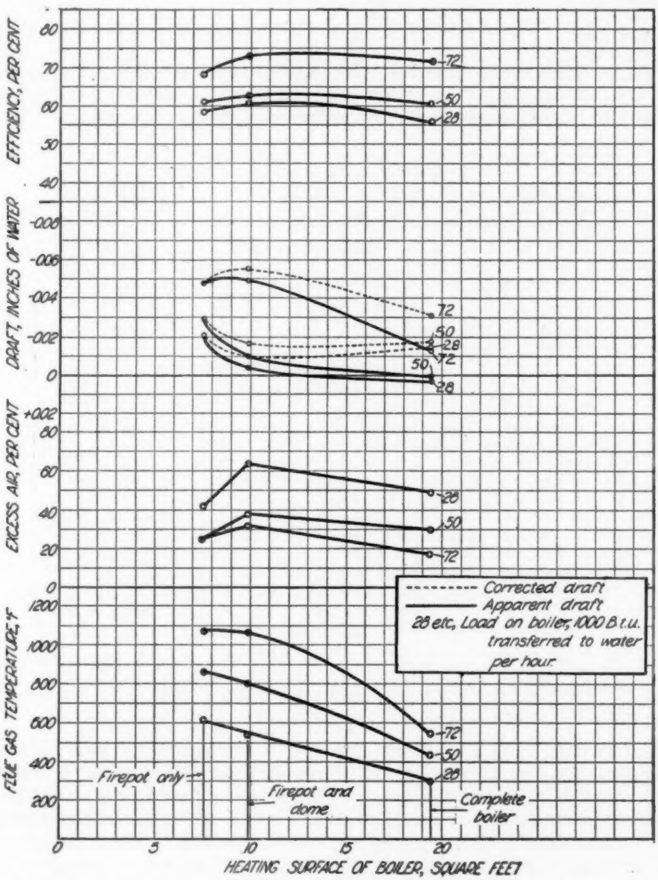


FIG. 11. CURVES SHOWING THE THERMAL EFFICIENCIES BASED ON FUEL GASIFIED, THE DRAFT, THE TEMPERATURE AND EXCESS AIR CONTENTS OF FLUE GASES FOR ANTHRACITE. PORTS IN AIR-DOOR CLOSED, FOR THREE LOADS, FROM THREE SERIES OF TESTS

some secondary air through the firing door partly to supply the deficiency passing around the sides of the fuel bed and partly to create the necessary turbulent flow for proper ignition at high temperature points within the combustion space of the fire-pot.

It should be noted that owing to the better results obtained in this particular boiler by the spreading method of firing over the coking method of firing in the first two series of tests, that the present series was fired entirely by the spreading method. A complete description of this rather interesting condition having been given in the previous report, same will not be repeated herein.

Secondary Air

In the present series of coke tests the introduction of secondary air by means of the slots in the firing door at low rates showed little difference from the test with the slots closed, which probably resulted from the fact that the draft was not

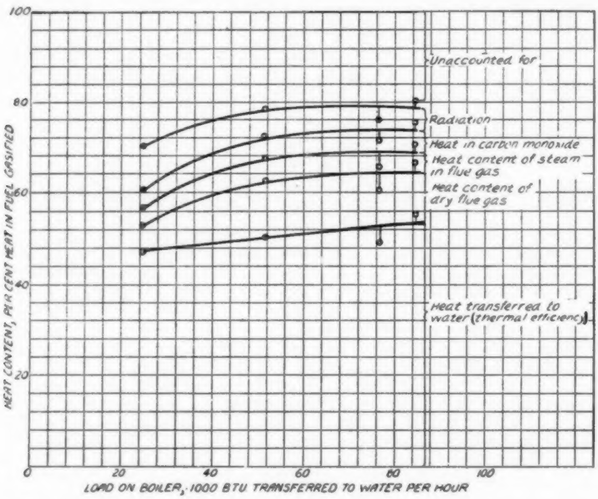


FIG. 12. HEAT ACCOUNT BASED ON FUEL GASIFIED, BITUMINOUS COAL. THIRD SERIES, COMPLETE BOILER. NO SECONDARY AIR, PORTS IN FIRE-DOOR CLOSED

sufficient to draw in an appreciable quantity of air through the slots in the door when open. At the higher rate the efficiency with the slots closed was about 2 per cent greater than with the slots open. The anthracite tests having been extended up to near the maximum rating of the boiler for the first time in the three series offers a particularly interesting study in the introduction of secondary air. Previous series showed decreases in efficiencies with the slot open as compared to the efficiencies with the slot closed, as did the present series with the exception of the special high rate tests 73 and 97. The order of such net efficiency decreases in the present series varied from 1.5 to 6.6 per cent of the total thermal efficiency of the boiler. However with the two special high rate tests the reverse condition proves true, *i. e.*, that test 97 with slots open showed 7.6 per cent greater thermal efficiency than test 73 with slots closed. This proves that when the ratings carried are high enough anthracite will show the same characteristics as bituminous shows both on the past and present series at a lower range of ratings. That is, efficiency is raised by the admission of secondary air at the higher ratings when the available com-

bustion volume is crowded with unmixed, partially burned gases and the efficiency is lowered when admitting secondary air at lowest ratings when a larger proportion of excess air enters from beneath the grates and when the fuel bed temperatures are lower.

Again referring to the moderate and low rate portions of the curves attention is particularly invited to the fact that the above discussion, together with the previous discussion under the sub-title of *Passage of free oxygen past the fuel bed,* applies to *average* efficiencies throughout the duration of the test as a whole and does not apply to intermittent periods where undoubtedly the efficiency can be increased by the introduction of a limited quantity of excess air for a limited period. The discussion

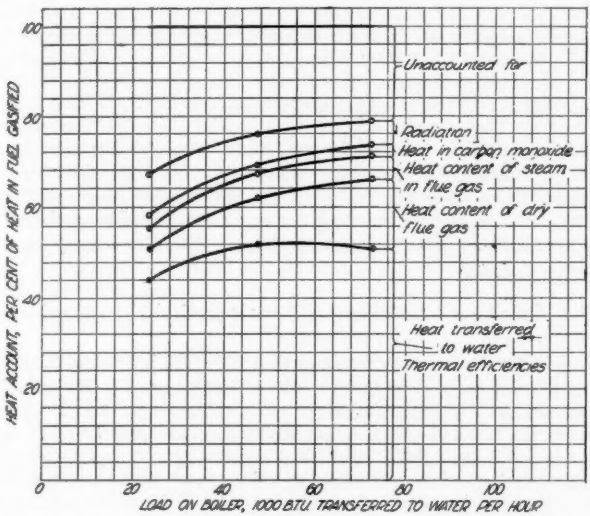


FIG. 13. HEAT ACCOUNT BASED ON FUEL GASIFIED, BITUMINOUS COAL. THIRD SERIES, COMPLETE BOILER. SECONDARY AIR PORTS IN FIRE-DOOR WIDE OPEN

so far is only given to prove that when burning coke or anthracite at low or moderate ratings leaving the slots in the firing door open *continuously* will result in a very appreciable net decrease in efficiency. If, however, limited quantities of secondary air are introduced while the combustible volatile is in process of distillation and only during such time as the exposed incandescent fuel bed provides sufficiently high temperatures to ignite same, then net increases of efficiency of operation may be expected and these increases in efficiency are clearly indicated by an examination of individual test results on the original detailed data sheets in this series of tests, as well as by detailed data sheets of the previous series which were reported in Tables 5, 6 and 7, of the previous report. These clearly show that during the first part of the firing period the secondary air drawn in does cut down the evaporation for as previously indicated the top of the charge of fuel is not then burning so that the secondary air merely serves to dilute the flue gases as the temperature of the mixture is below ignition temperature. During the latter part of the firing period

the top of the charge begins to burn, the temperature above the fuel bed increases until the distilled combustible gases ignite and very good combustion occurs with very little carbon monoxide escaping up the stack. During this part of the firing period the evaporation and efficiency goes up higher with slots open than when no secondary air is admitted. The average for the entire period, however, weighs clearly against the *continuous* admission of secondary air except at high ratings for the results indicate that net continuous efficiencies at low and moderate ratings, with slots closed, are higher by 1.5 per cent to 3.0 per cent in the case of anthracite and from 0 to 2 per cent in the case of coke than when operating with slots continuously open.

The natural gas tests of the present series show results in general parallel to those of the tests with coke and anthracite. Here, obviously, assuming the primary air to be properly proportioned in the first place for efficient combustion, secondary air can only result in lower stack temperatures and decreased efficiencies. The same trend of decrease in carbon monoxide losses with increasing load up to the rating of maximum efficiency is noticeable in the data of Table 2 as with the coke and anthracite. The presence of excess oxygen in the first three low rate tests, numbers 65, 66 and 67, is explained by the fact that the primary air supply was not readjusted for maximum efficiency in each test, having been left the same for all tests in an endeavor to more nearly simulate practical service conditions. Obviously, the proper primary air setting for high rates would result in an excess of primary air for the lower rates and this condition is demonstrated by the data.

Flue Gas Temperatures

The effect of the three intermediate sections is particularly noticeable on the flue gas temperatures.

Reference to the flue gas temperature curves on the bottom of Fig. 6 indicates that on the present series of tests with coke at low ratings (28,000 B.t.u. transfer) these temperatures were approximately 165 deg. lower on the complete boiler than with the fire-pot and dome structure and approximately 390 deg. lower than with the fire-pot only. With increasing ratings these temperature differences become larger so that at 72,000 B.t.u. transfer the complete boiler flue gas temperature is about 400 deg. lower than with fire-pot and dome and about 600 degrees lower than with fire-pot only. As would be expected decrease of flue gas temperature is greater with the slots in the firing door open than when same are closed, and secondary air drawn in serving to dilute the gases of the combustion chamber.

During the previous anthracite series, using fire-pot and dome, with the slots closed, the flue gas temperature increased with increasing load from 435 to 1076 deg. fahr. During the present series they increased with the load from 297 to 587 deg. fahr. As will be seen from the flue gas temperature curves of Fig. 10 this represents a decrease in flue gas temperature of from 44 per cent in the case of a low load such as 28,000 B.t.u. transferred per hour to as high as 50 per cent at a load of 72,000 B.t.u. transferred per hour. It is noticeable that curve *c* of tests with complete boiler tends to diverge with increasing loads from curve *b* of tests of boiler consisting of fire-pot and dome while the curve *a* of tests with fire-pot only tends to converge with increasing loads with curve *b* of tests with fire-pot and dome. A somewhat analogous condition also is noticeable in the coke curves of Fig. 6. This clearly indicates that in the previous series of tests with limited flue gas travel combustion continued up to and perhaps beyond the base of the stack so that the maximum flue gas temperature was probably somewhere beyond the point at

which it was measured, about 4 in. above the point where the flue joined the boiler. With the increased length of gas travel when the three sections were added, however, combustion was completed within the furnace itself resulting in the dropping of the flue gas temperature curves of the present series away from those of the previous series.

The flue gas temperatures on the three series of bituminous tests are shown on the bottom of Fig. 14. At 28,000 B.t.u., with slots closed, the flue gas temperature is 110 deg. lower with the fire-pot and dome than with the fire-pot only and 430 deg. lower with the complete boiler than with the fire-pot only. At 60,000 B.t.u. transfer rate, with slots closed, the fire-pot and dome temperature is about 75 deg. lower than that with the fire-pot only and about 490 deg. lower in the complete boiler than with the fire-pot only. The same tendency towards convergence of the temperature curves in the first two series and divergence in the present series is noticeable here as was noticed in the coke and anthracite tests, again clearly indicating the progress of combustion beyond the entrance of the stack until sufficient sections were added to prevent. Similar tendencies are also noticeable in the natural gas curves of Fig. 16, the decrease of gas temperature from the high temperatures recorded with fire-pot only being at 28,000 B.t.u., 160 deg. with fire-pot and dome and 510 deg. with the complete boiler. At 60,000 B.t.u. the decrease with fire-pot and dome was 140 deg., while with the complete boiler it attained the rather astonishing difference of 800 deg. It is of utmost importance therefore when burning natural gas of high B.t.u. value that sufficient length of gas travel be provided to permit complete combustion and absorption of the heat so generated prior to arrival at the smoke outlet.

Drafts

One of the incidental objectives tentatively proposed in the previous series of tests was to ascertain effect of drafts on the addition of sections. The series of stack draft curves shown on Figs. 6, 10 and 14 show for the coke, anthracite, and bituminous tests, respectively, the trend of drafts which might be expected, *i. e.*, a gradual increase of draft with increase of load and increasing fuel bed temperatures. The curves of Figs. 7, 11 and 15, plotted from points taken from the preceding curves at 28,000, 50,000 and 72,000 B.t.u. heat transfer, all indicate a downward trend of draft with increase of heating surface. This is doubtless due to the lower excess air content and the more complete combustion made possible when sufficient combustion volume and length of gas travel was provided in the complete boiler. Thus there was a lesser volume of gases to be moved at a given combustion rate, naturally requiring less draft to move them and this effect apparently more than outweighed the increase of frictional resistance in the flues of the sections.

The stack draft curves do not exactly show the true trend of draft increase for which reason it will be noted that corrected drafts have also been computed and sketched in the form of dotted curves. These take into consideration the differences of vertical height and of gas density between the hot gas inside and the cold air outside existent with the three different furnace set-ups considered.

Since the density of air is 0.074 lb. per cu. ft. at 75 deg. fahr. (which we may take as average outside air temperature) and 0.192 in. the height of water column corresponding to a pressure of 1 lb. per sq. ft., then the correction per foot to be applied for gases inside and outside of equal density would be the product of 0.074×0.192 or 0.014 in. of water per foot of height. But the gas inside was at a mean temperature of about 1300 deg. fahr. so that assuming the density of the gas inside

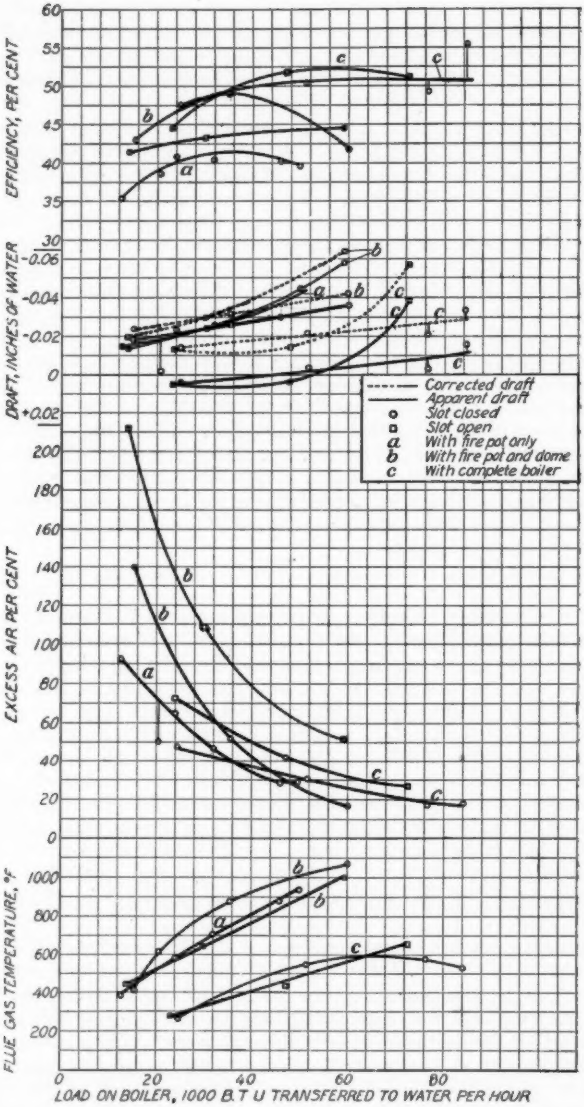


FIG. 14. CURVES SHOWING THE THERMAL EFFICIENCIES BASED ON FUEL GASIFIED, DRAFT, TEMPERATURES AND EXCESS AIR CONTENT OF FLUE GASES FOR BITUMINOUS COAL

at normal temperature and pressure to have been 4 per cent greater than that of air the ratio of the density of the gas at 1300 deg. fahr. to the air at 75 deg. fahr. was $\dfrac{(460 + 75) \times 1.04}{1300 + 460} = 0.3$ approximately. As a 1 ft. column of air exerts a pressure of 0.014 in. of water, a 1 ft. column of hot gas would exert a counter pressure of 0.3 ×

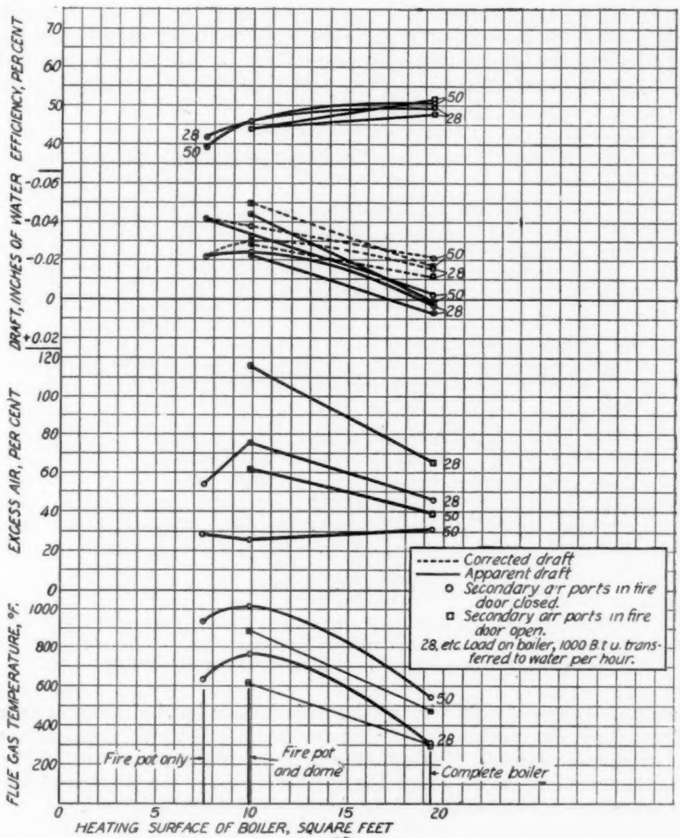


FIG. 15. CURVES SHOWING THE THERMAL EFFICIENCIES BASED ON FUEL GASIFIED, THE DRAFTS, TEMPERATURES AND EXCESS AIR CONTENTS OF THE FLUE GASES, WITH BITUMINOUS COAL, FOR THREE SERIES OF TESTS

0.014 or 0.004 in. of water, therefore the net pressure correction per foot of height added to the fire-pot was 0.014 − 0.004 = 0.01 in. of water.

Application of this correction factor to these three curves results in the corrected (dotted) draft curves which indicate the true shape of draft tendency with the addition of heating surface. They still, however, do not represent the total net drafts

used to force the gas through the fuel bed, to overcome the frictional resistance through the fire-pot, sections and dome, to increase the velocity of the gases, and to support the approximate 2.5-ft. column of hot gases in the fire-pot itself. To do this an additional correction factor of approximately 0.025, similarly figured for the difference of levels at which the drafts were taken in the ash-pit and in the base of the stack, should have been applied which would merely serve to raise the three curves by the amount indicated without changing their form or relative position. This is mentioned, however, to explain the paradox of seemingly higher drafts in the ash-pit at the lower ratings than were recorded at the stack, as well as the positive pressures recorded in the stack in some of these low rate tests. The same

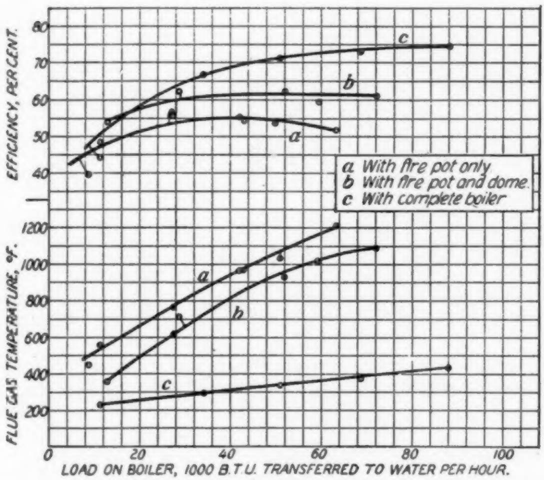


FIG. 16. CURVES SHOWING THE THERMAL EFFICIENCIES AND FLUE GAS TEMPERATURE WITH NATURAL GAS. SLOTS IN FURNACE DOOR CLOSED, ON THREE SERIES OF TESTS

phenomenon may be observed in any domestic furnace by closing the damper in the flue and opening the ash-pit door, when gases will leave the furnace through any openings near the top. Evidently the pressure inside is greater than that outside at the same level at the top of the furnace, although the pressure of the air inside the ash-pit is less than that of the air outside at the ash-pit level.

Description of Equipment Tested

The boiler used in these tests was a small circular cast iron boiler presented to the Bureau of Mines by the United States Radiator Corp. Its principal dimensions are shown in the assembly drawing, Fig. 1. More detailed dimensions are given in the previous report.

Fig. 2 shows the boiler as it was operated during the gas tests.

Fig. 3 shows the boiler assembled for the solid fuel tests. For test purposes it was necessary to resort to separators in order that dry steam might be produced at

atmospheric pressure without priming. Under normal operating conditions at 2 lb. pressure, as recommended by the manufacturer, it is probable that a separator would not have been necessary. To secure accurate test results, however, the production of dry steam is a prime essential and this, coupled with the greater ease of manipulation on test with atmospheric pressure, made it desirable to operate in the manner indicated.

It will be noticed from Figs. 2 and 3 that a different separator was used on the gas tests from that used on the solid fuel tests. This was changed because the limited capacity and the type of separator used on the gas tests which were run first developed a tendency to prime at the highest rates. The separator was connected so that any separated water was returned automatically to the boiler. In addition to separated water there was also returned to the boiler condensate resulting from steam condensed to supply radiation and convection losses from the separator. These losses are, of course, not inconsiderable and consequently they serve to decrease the efficiency of the boiler as determined by the tests. Comparative values only, however, as indicated by the original objects of the tests were sought, hence the accuracy of the comparative values obtained is in no way impaired by the somewhat artificial efficiency figures indicated.

Method of Conducting Tests and Computing Results

The boiler was mounted on a platform scale and the gage glass calibrated to show the weight of water in the boiler. Thus it was possible to weigh the amount of fuel and ash in the fire-pot at any time and so determine the rate of combustion during any part of the test as well as the total weight of fuel gasified during the whole trial. The fuel was fired in charges of 30 lb. each (with the exception of two special high rate anthracite tests, numbers 73 and 97). The charging rate of 30 lb. was selected on all of the regular tests because it was the greatest amount of coke (the bulkiest fuel used) that could be charged at one time and it was desired to fire all fuels at all rates in a precisely similar manner. The length of firing interval was selected as inversely proportional to the rate of combustion and varied from 3 to 15 hrs. On the anthracite tests (excepting tests 73 and 97) a fixed weight of 210 lb. of fuel per test was charged. As it was possible to mount the entire equipment on platform scales and to accurately weigh same (owing, however, to the relatively light weight and small volume of the boiler under test), it was decided that the lengthy low rate tests usually demanded to minimize possible inaccuracies in gaging the fuel bed on starting and completing the tests were unnecessary. Hence, subsequent tests on coke and bituminous coals were cut down to as near an approximation of the 16-hr. full rate tests specified in the low pressure boiler code as the 30-lb. charging rate decided upon would permit. With the high order of accuracy with which it was possible to weight this small boiler and with the very careful sampling and analysis which was done, it is believed that the results on these tests will prove fully as accurate, as the more usual longer tests demanded where the starting and stopping contents of the fire-pot must be personally estimated by the observer.

The rate of combustion was kept as nearly constant as possible throughout the trial by manipulating the draft, removing ash and, when necessary, breaking up the bituminous fuel bed. In order to as closely as possible simulate practical operating conditions, as little attention as possible was given the fuel bed. Amount of attention given in the various tests, averaged to a common 8-hour basis, is shown in Table 1.

Fuel was charged when the weight of the boiler indicated that the previous charge had been consumed. No attempt was made at any time during the tests to manipu-

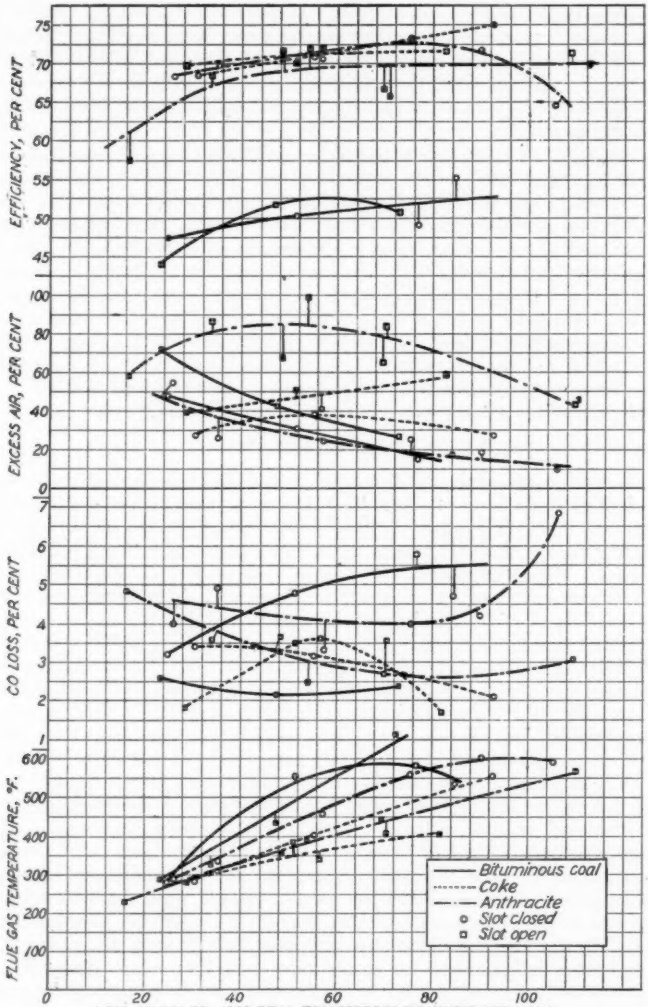


FIG. 17. CURVES SHOWING THE THERMAL EFFICIENCIES BASED ON FUEL GASIFIED, FLUE GAS TEMPERATURES, CARBON MONOXIDE LOSS AND EXCESS AIR CONTENT OF FLUE GAS, FOR VARIOUS FUELS USING COMPLETE BOILER

late the fire or draft temperatures to improve the efficiency. The attention given, therefore, as indicated in Table 1, was that required to maintain a constant rate of evaporation.

TABLE 1. ATTENTION TO BOILER ON 8-HOUR BASIS

Kind of fuel	Test no.	Fuel rate, lbs. per hr.	No. times shaking	No. times poking	No. times changing draft
		Slide closed			
Anthracite	63	3.2	1	0	2
Anthracite	65	7.1	2	0	2
Anthracite	67	10.8	3	0	2
Anthracite	68	9.2	3	0	2
Anthracite	69	4.8	2	0	3
Anthracite	73	14.9	5	0	2
		Slide open			
Anthracite	64	2.3	1	0	1
Anthracite	65A	9.3	2	0	3
Anthracite	66A	6.6	2	0	2
Anthracite	70	10.3	6	0	2
Anthracite	71	6.6	2	0	3
Anthracite	72	4.4	1	0	4
Anthracite	97	13.9	2	0	1
		Slide closed			
Bituminous coal	85	4.3	0	0	1
Bituminous coal	89	11.4	1	2	0
Bituminous coal	87A	8.0	3	4	0
Bituminous coal	89A	12.0	1	5	0
		Slide open			
Bituminous coal	86	4.3	1	0	1
Bituminous coal	88	7.3	2	0	2
Bituminous coal	90	10.9	1	3	0
		Slide closed			
Coke	91	10.0	1	0	0
Coke	92	6.4	3	0	1
Coke	93	3.7	1	0	3
		Slide open			
Coke	94	3.4	1	0	3
Coke	95	6.2	2	0	2
Coke	95A	6.5	3	0	2
Coke	96A	9.6	3	0	4

From the total weights of fuel fired and refuse removed, the change in weight of the contents of the fire-pot during the trial, and the composition and calorific value of the fuel charged and refuse removed, the total heat in the fuel gasified was calculated. It was assumed that all the ash in the fuel charged was either removed with the refuse from the ash-pit or remained in the fire-pot and that the fire-pot contained only ash and carbon. The first assumption, that all the ash in the coal was removed with the refuse or remained in the fire-pot, is not strictly accurate, since it is well known that some particles of ash are always carried up the flues with the flue gases; but it is unlikely that, at low rates of combustion, the quantity of ash carried up the stack is large enough to cause the method used to be greatly in error. The latter assumption, that the fire-pot contained only ash and carbon, was probably approximately true, since the trial was started and stopped just before it became necessary to charge the fuel, so that the bulk of the volatile contents of the fuel had been driven out of the fire-pot.

The results of these tests are shown both in tabular form and graphically. Table 2 shows the results of the anthracite and coke tests of the third series. Table 3 presents the results of the natural gas tests.

RESULTS OF TESTS, THIRD SERIES,

Item No.	Particulars of observation	Units	Anthracite								
	Position of fire door slide		Closed	Open	Closed	Open	Closed	Open	Closed	Open	Closed
1	Number of test		63	64	69	72	65	64A	68	71	67
2	Duration of test	Hours	47.0	92.0	43.5	47.0	33.5	32.5	24.0	31.0	19.0
	FUEL										
3	Calorific value per lb. as fired	B.t.u.	12310	12290	12030	12300	12460	11980	12410	12400	12080
	Ultimate analysis as fired										
4	Hydrogen	Per cent	2.8	2.8	2.7	2.7	2.7	3.2	2.7	2.8	2.7
5	Carbon	" "	76.0	75.6	74.7	76.1	76.8	73.5	76.6	76.7	74.6
6	Nitrogen	" "	0.9	0.9	0.9	0.9	0.9	0.9	0.9	0.9	0.9
7	Oxygen	" "	5.8	5.1	4.9	4.5	4.7	9.3	4.6	5.2	5.1
8	Sulphur	" "	0.9	1.1	1.1	0.9	1.1	1.0	1.0	1.0	0.9
9	Ash	" "	13.6	14.5	15.7	14.8	13.8	12.1	14.2	13.4	15.8
	Proximate analysis as fired										
10	Moisture	Per cent	4.0	3.3	3.0	2.7	2.7	8.0	2.6	3.3	3.3
11	Volatile matter	" "	7.4	6.3	6.1	5.9	6.4	6.2	6.3	5.0	6.8
12	Fixed carbon	" "	75.0	75.9	75.2	76.6	77.1	73.7	76.9	78.3	74.1
13	Ash	" "	13.6	14.5	15.7	14.8	13.8	12.1	14.2	13.4	15.8
14	Ratio, fixed carbon - volatile matter		10.1	12.1	12.3	13.0	12.0	11.9	12.2	15.7	10.9
15	Total charged during test	Lb.	210	210	210	210	210	210	210	210	211
	FUEL BED										
	Mass of fire-pot contents										
16	Increase of mass	Lb.	-2	-5	2	2	-29	-3	-10	4	6
17	Increase of carbon content of fire-pot	"	-8.1	-14.2	3.6	-0.7	33.0	0.2	-5.7	5.5	-2.4
18	Increase of ash content of fire-pot	"	6.1	9.2	-1.6	2.7	4.0	-3.2	-4.3	-1.5	8.4
	ASH AND REFUSE										
19	Carbonaceous matter in ash and refuse removed	Per cent	31.9	39.4	35.7	38.9	51.0	43.9	32.4	31.1	36.0
20	Total ash and refuse removed, per cent of fuel used	Per cent	15.6	16.3	26.0	22.4	21.3	27.9	22.3	20.9	19.0
	DRAFT										
21	Ash pit	In. water	-.011	-.008	-.004	-.001	.000	.000	.000	.000	.000
22	Furnace	" "	-.012	-.003	-.011	-.009	-.015	-.011	-.034	-.018	-.033
23	Stack	" "	+.004	+.009	+.010	+.003	-.002	-.002	-.025	-.015	-.033
	FEED WATER										
24	Total to boiler	Lb.	1559	1331	1360	1480	1708	1399	1600	1495	1518
25	Temperature	Deg. F.	65	65	57	60	58	62	53	56	54
26	Evaporation per lb. fuel used	Lb.	7.35	6.19	6.54	6.83	7.14	6.57	7.27	7.26	7.40
	FLUE GASES AND AIR										
27	Carbon dioxide	Per cent	11.7	10.8	14.0	9.7	14.4	10.7	14.4	9.3	15.2
28	Oxygen	" "	7.9	8.3	5.1	10.1	4.5	8.9	4.8	10.6	3.9
29	Carbon monoxide	" "	0.8	0.9	1.2	0.6	0.8	0.7	1.0	0.4	1.1
30	Nitrogen (by difference)	" "	79.6	80.0	79.7	79.6	80.2	79.7	79.8	79.7	79.8
31	Excess air content	" "	55	59	27	87	25	68	26	98	19
32	Temperature of flue gases	Deg. F.	297	230	332	329	459	386	561	397	602
33	Temperature of air entering ash pit	" "	77	75	75	74	74	77	74	74	72
	RATES										
34	Heat transferred to water per hour	1000 B.t.u.	26	16	36	34	57	48	75	54	90
35	Heat transferred to water per sq. ft. of heating surface per hour	B.t.u.	1340	830	1810	1750	2960	2490	3880	2800	4460
36	Fuel used per hr.	Lb.	3.2	2.3	4.8	4.4	7.1	6.6	9.2	6.6	10.8
37	Fuel used per hr. per sq. ft. grate area	"	1.5	1.1	2.2	2.0	3.2	3.0	4.2	3.0	4.9
	HEAT IN FUEL (per cent of heat in fuel charged)										
38	Increase in heat content of fuel bed	Per cent	-4.6	-8.0	2.2	-0.4	-18.3	0.1	-3.2	3.1	-1.4
39	Total heat available	" "	104.6	108.0	97.8	100.4	118.3	99.9	103.2	96.9	101.4
40	Heat in combustible in ash	" "	5.9	7.8	11.1	10.2	14.4	13.0	8.9	7.5	8.0
41	Heat in fuel gasified (Item 39 - Item 40)	" "	98.7	100.2	86.7	90.2	103.9	86.9	94.3	89.4	93.4
	HEAT BALANCE										
42	Heat transferred to water (Thermal efficiency)	Per cent	68.3	57.5	69.9	68.4	70.7	71.7	73.5	72.3	72.0
43	Heat carried away by steam in flue gas	" "	2.4	2.3	2.7	2.5	2.3	3.3	2.6	2.0	2.8
44	Heat carried away by dry flue gas	" "	7.0	4.9	6.6	9.9	9.6	8.9	12.5	12.4	12.7
45	Loss due to carbon monoxide	" ?	4.0	4.8	4.9	3.6	3.3	3.7	4.0	2.5	4.2
46	Loss due to radiation and convection	" "	13.4	16.4	11.0	11.1	8.2	9.2	7.4	8.7	6.6
47	Undetermined losses	" "	4.9	14.1	4.9	4.5	5.9	3.2	0.0	1.3	1.7
48	Total of items 42 to 47 or heat in fuel gasified	Per cent	100.0	100.0	100.0	100.0	100.0	100.0	100.0	100.0	100.0
	THERMAL EFFICIENCIES										
49	Grate	Per cent	94.3	92.8	88.7	89.9	87.8	87.0	91.4	92.3	92.1
50	Boiler and furnace (item 42)	" "	68.3	57.5	69.9	68.4	70.7	71.7	73.5	72.3	72.0
51	Boiler, furnace and grate	" "	64.4	53.3	62.0	61.5	62.0	62.4	67.2	66.7	66.3

The exact procedure in calculating the fuel gasified is here illustrated by concrete example from test 72 as follows:

During this test 210 lb. of anthracite was fired containing 14.8 per cent of ash. The weight of ash and refuse removed weighed 46.5 lb. and contained 61.1 per cent ash or 28.4 lb. of actual ash. The weight of ash charged was 210 × 14.8 or 31.1 lb. Since 31.1 lb. of ash were charged and only 28.4 lb. of ash were removed, the fire-pot contained 31.1 − 28.4 or 2.7 lb. more ash at the end of the test than at the beginning. The weight of the contents of the fire-pot increased by 2 lb. (Item 16, Table 2) and since it contained 2.7 lb. more ash than when the test started it must have lost 2.7 − 2 or 0.7 lb. of combustible. (See Item 17, Table 2.) Thus the total heat available for use was the sum of the total heat contents of the anthracite charged and of the 0.7 lb. depletion of the combustible contents of the fire-pot. Assuming

ANTHRACITE, BITUMINOUS COAL AND COKE

Bituminous coal											Coke						
Open	Open	Closed	Open	Closed	Open	Closed	Open	Closed	Closed	Open	Closed	Open	Closed	Open	Open	Closed	Open
68A	70	73	97	85	86	87A	88	89A	89	90	93	94	92	95	95A	91	94A
22.5	20.5	20.0	17.3	20.0	27.5	15.0	16.4	10.0	10.5	11.0	16.0	17.5	14.0	14.5	13.6	12.0	12.5
12810	12030	12750	13040	13520	13470	13670	13630	13670	13550	13770	12680	12680	12700	12570	12680	12370	12590
2.8	2.6	2.9	2.5	5.5	5.5	5.6	5.6	5.6	5.6	5.6	1.0	1.0	1.0	1.0	1.7	0.9	1.0
79.0	74.7	78.8	81.0	75.6	75.5	76.9	76.7	76.9	76.4	77.1	84.9	84.9	85.3	84.7	84.9	83.8	85.4
0.9	0.9	0.9	0.8	1.4	1.4	1.5	1.5	1.5	1.4	1.5	1.3	1.3	1.3	1.3	1.3	1.3	1.3
3.5	4.7	4.1	4.0	8.7	8.8	9.0	8.7	9.0	8.9	9.1	0.9	0.9	0.7	0.7	0.7	0.6	0.2
0.8	0.9	0.9	0.7	1.6	1.6	1.3	1.4	1.3	1.4	1.3	0.8	0.8	0.8	1.0	0.8	0.9	1.1
13.0	16.2	12.4	11.0	7.2	7.2	5.7	6.1	5.7	6.3	5.4	11.1	11.1	10.9	11.3	10.6	12.5	11.0
1.9	2.8	2.6	1.6	2.5	2.6	2.7	2.4	2.7	2.6	2.8	0.9	0.9	0.7	0.7	0.7	0.7	0.2
5.5	5.6	5.2	5.4	38.1	38.2	38.8	39.0	38.8	39.7	39.9	3.5	3.5	3.9	2.7	1.9	2.3	1.7
79.6	75.4	79.8	82.0	52.2	52.0	52.8	52.5	52.8	51.4	51.9	84.5	84.5	84.5	85.3	86.8	84.5	87.1
13.0	16.2	12.4	11.0	7.2	7.2	5.7	6.1	5.7	6.3	5.4	11.1	11.1	10.9	11.3	10.6	12.5	11.0
14.5	13.8	15.3	15.2	1.4	1.4	1.4	1.3	1.4	1.3	1.3	24.1	24.1	21.7	31.6	45.7	21.7	51.2
210	210	420	240	120	120	120	120	120	120	120	60	60	90	90	90	120	120
0	-2	+3	0	+1	+1	0	0	0	0	0	0	+1	0	0	0	0	0
-0.4	-6.7	-1.5	-2.9	1.0	0.3	-1.6	+0.4	-1.6	-2.1	+0.4	-0.5	-0.3	-0.8	-0.3	-1.2	-2.8	-3.6
0.4	4.7	4.5	2.9	0.1	+0.7	+1.6	-0.4	1.6	2.1	-0.4	+0.5	+0.3	+0.8	+0.3	+1.2	+2.8	+3.6
47.3	50.3	52.2	60.9	61.3	52.9	61.8	53.9	54.8	39.5	42.4	38.1	36.7	32.8	33.8	36.0	33.7	46.9
34.3	27.8	34.0	25.0	15.1	14.3	11.7	14.2	9.6	7.5	10.0	16.7	17.0	15.0	16.7	14.4	12.9	18.0
.000	.000	.000	-.002	-.011	-.009	-.004	-.005	-.003	-.004	-.003	-.012	-.009	-.004	-.004	-.008	-.006	-.004
-.018	-.026	-.036	-.028	-.011	-.006	-.005	-.009	-.020	-.019	-.017	-.006	-.002	-.004	-.002	-.006	-.015	-.012
-.025	-.037	-.049	-.045	+.004	+.005	-.009	+.003	-.020	-.017	-.039	+.010	+.014	+.008	+.008	+.010	-.012	-.085
1396	1276	2642	1686	628	584	693	696	686	795	716	438	447	691	664	701	998	817
59	54	62	60	68	69	65	65	64	64	64	66	67	64	65	57	64	58
6.65	6.02	6.33	7.03	5.28	4.91	5.78	5.80	5.72	6.63	5.97	7.30	7.58	7.68	7.38	7.79	8.38	7.64
10.9	9.9	15.3	13.0	11.6	10.1	13.0	12.0	14.0	14.5	13.2	15.2	14.0	14.0	12.9	13.8	15.7	12.3
8.6	9.9	2.9	6.4	6.9	8.9	5.0	6.3	3.2	3.3	4.6	4.6	6.0	5.9	7.1	6.1	4.5	7.8
0.5	0.6	1.9	0.7	0.7	0.5	1.2	0.5	1.6	1.3	0.6	0.8	0.4	0.7	0.7	0.8	0.5	0.3
80.0	79.6	79.9	79.9	80.8	80.5	80.8	81.2	81.2	80.9	81.6	79.4	79.6	79.4	79.3	79.3	79.3	79.6
65	64	10	44	48	72	31	42	18	18	27	28	40	39	51	41	27	59
445	407	587	567	276	287	552	640	585	529	663	283	281	405	385	343	553	407
77	74	78	68	69	68	71	65	72	69	66	69	71	74	72	66	69	68
70	70	106	110	25	24	52	47	77	86	73	31	28	55	51	57	93	82
3590	3620	5450	5656	1289	1220	2663	2442	3954	4364	3754	1575	1464	2846	2638	2947	4798	4252
9.3	10.3	14.9	13.9	4.3	4.3	8.0	7.3	12.0	11.8	10.9	3.8	3.4	6.4	6.2	6.5	10.0	9.6
4.2	4.7	6.8	6.3	1.9	2.0	3.6	3.3	5.5	5.2	5.0	1.7	1.6	2.9	2.8	3.0	4.6	4.4
-0.2	-3.8	-0.4	1.4	0.9	0.3	1.3	0.4	-1.4	-1.9	0.4	-0.9	-0.7	-1.0	-0.4	-1.6	-2.7	-3.5
100.2	103.8	100.4	101.4	99.1	99.7	101.3	99.6	101.4	101.9	99.6	100.9	100.7	101.0	99.6	101.6	102.7	103.5
13.0	17.1	14.2	17.0	8.3	8.0	7.7	8.1	5.6	3.2	4.5	7.2	7.0	5.6	6.5	6.0	5.0	8.1
87.2	86.7	86.2	84.4	90.8	91.7	93.7	91.5	95.8	98.7	95.1	93.7	93.7	95.4	93.1	95.6	97.7	95.4
66.9	65.7	64.1	71.7	47.5	44.0	50.4	52.0	48.8	55.4	51.0	68.6	69.7	70.9	70.0	72.3	75.0	71.5
2.7	2.7	3.1	2.6	4.6	4.6	5.0	4.9	5.0	4.8	5.1	0.8	0.8	0.9	0.9	1.5	1.0	0.9
12.1	11.9	10.7	13.6	5.6	6.9	12.1	10.2	11.6	10.7	15.1	5.6	6.2	9.6	10.0	8.0	13.0	11.2
2.7	3.6	6.8	3.1	3.2	2.6	4.8	2.2	5.8	4.7	2.4	3.4	1.8	3.2	3.5	3.7	2.1	1.7
6.9	6.6	5.5	6.1	9.6	9.3	6.3	6.7	4.8	5.2	5.2	11.9	12.8	8.4	8.6	8.4	6.8	6.9
8.7	9.3	9.8	2.9	29.5	32.6	21.4	24.0	24.0	19.2	21.2	9.7	8.7	7.0	7.0	6.1	2.1	7.8
100.0	100.0	100.0	100.0	100.0	100.0	100.0	100.0	100.0	100.0	100.0	100.0	100.0	100.0	100.0	100.0	100.0	100.0
87.0	83.5	85.8	83.3	91.7	91.9	92.4	91.8	94.5	96.9	95.5	92.8	93.1	94.5	93.5	94.1	95.1	92.1
66.9	65.7	64.1	71.7	47.5	44.0	50.4	52.0	48.8	55.4	51.0	68.6	69.7	70.9	70.0	72.3	75.0	71.5
59.2	54.9	55.1	59.7	43.5	40.4	46.6	47.7	46.1	53.7	49.7	63.7	64.9	67.0	65.4	68.1	71.4	65.9

the 0.7 lb. of combustible to be carbon having a calorific value of 14,540 B.t.u. per lb. then since the anthracite charged had a calorific value of 12,300 B.t.u. per lb., the total heat available was (210 × 12,300) + (0.7 × 14,540 B.t.u.) or $\left(1 + \frac{0.7 \times 14{,}540}{210 \times 12{,}300}\right) \times 100$ per cent of the heat in the fuel charged which is shown in Item 39 to be 100.4 per cent. In a similar manner the heat in the combustible in the refuse removed may be shown to represent 10.2 per cent. The total heat in the fuel gasified then is the difference between 100.4 per cent and 10.2 per cent or 90.2 per cent of the heat in the fuel actually charged. (See Item 41.)

The flue gas temperature was measured by means of an exposed copper-constantan thermocouple and potentiometer. The hot junction of the thermocouple

was placed in the center of the flue leading from the boiler and about 4 in. above the point where the flue joined the boiler.

The sample of flue-gas for analysis was drawn from a point near the hot junction of the thermocouple. It was drawn into a bottle at a steady rate for about a half-hour, and then analyzed in an ordinary Orsat apparatus.

The draft was measured in the stack about 4 in. above the boiler, in the furnace at about the level of the bottom of the fire-door, and in the ash-pit, using inclined U-tubes filled with gasoline and calibrated to read directly in thousandths of an inch of water.

The water fed to the boiler was weighed, and its temperature was taken by means of a mercury thermometer. It was fed to the boiler intermittently, and the level of the water in the boiler was read every half-hour. It was only by feeding the water in fairly large quantities just before reading the water level in the gage-glass that the priming subsided sufficiently to enable the correct level of the water in the gage-glass to be read. Closing the steam line momentarily also aided in checking the fluctuation in the gage-glass. The steam passed from the boiler to the separator when the water passed back to the boiler and the steam directly to the atmosphere. In computing the results, the steam was assumed to be dry as ample separator capacity was provided to insure this. The pressure in the boiler was practically that of the atmosphere throughout. The steam temperature was measured by a mercury thermometer.

The radiation and convection loss from the outside surface of the boiler and steam separator was determined by maintaining the steam boiler at 212 deg. fahr. by means of an electric boiler attached to the main boiler and separator through the drain-valve thus using the main boiler as a radiator. The steam outlet on the main boiler was closed. The electric current was adjusted so that the steam temperature of the main boiler remained constant at 212 deg. fahr.; and therefore the input of electrical energy was dissipated by the outer surface of the two boilers and the separator. By experiment, the radiation and convection loss of the electric boiler was found so that the actual amount of energy dissipated at the outer surfaces of the main boiler and separator might be separated from the total electrical energy input. The total energy input was measured by means of an ammeter and voltmeter and for the boiler as used in the third series amounted to 3806 B.t.u. per hour. This loss represents the radiation and convection loss at zero load only and it was necessary to estimate it for the higher ratings where increased temperatures in the fire-pot would undoubtedly result in increased radiation and convection losses through the ash-pit and other parts not thoroughly insulated or cooled by the water in the boiler. This was done by plotting the total unaccounted-for losses in B.t.u. per hour against the rate of evaporation for coke, anthracite and natural gas, for which fuels the losses due to undetermined combustible gases in the flue gases should be small, and then drawing a line from the loss at zero load which seemed to represent the probable radiation and convection loss. The loss was so determined as to give the minimum radiation and convection loss, and the same loss for the same rating was used for all fuels, which undoubtedly involved some error.

The only results computed in a somewhat unusual manner are those already referred to; the remaining results were computed in the usual manner and need not be explained here.

All observations were made every 30 min.

The heat loss caused by carbon monoxide throughout the present series is seen to be comparatively low, ranging about the same or less for the completely as-

TABLE 3. RESULTS OF NATURAL GAS TESTS WITH COMPLETE BOILER

Item no.	Particulars of observation	Units	Complete boiler				
1	Number of test		67	66	65	64	63
2	Duration of test	Hours	2.5	3.0	3.0	3.0	3.0
	FUEL						
3	Calorific value per cu. ft. at 32° F. and 14.7 lb. per sq. in.	B.t.u.	1105	1105	1105	1105	1105
	Fuel gas analysis, by volume						
4	Methane	Per cent	88.9	88.9	88.9	88.9	88.9
5	Ethane	Per cent	8.5	8.5	8.5	8.5	8.5
6	Nitrogen (by difference)	Per cent	2.6	2.6	2.6	2.6	2.6
7	Gas pressure at burner	In. water	6.0	5.8	6.5	9.9	6.0
8	Total gas burned during test (at 32° F. and 14.7 lb. per sq. in.)	In. water	60.5	153.0	224.0	297.0	364.0
	DRAFT						
9	Under burner	In. water	−.016	−.022	−.023	−.020	−.010
10	Stack	In. water	+.006	+.001	+.001	+.001	+.006
	FEED WATER						
11	Total to boiler	Lb.	24	91	140	189	238
12	Temperature	Deg. F.	77	73	79	80	78
	FLUE GASES AND AIR						
13	Carbon dioxide	Per cent	7.3	8.8	10.3	10.3	11.3
14	Oxygen	Per cent	7.2	5.2	2.0	0.6	0.5
15	Carbon monoxide	Per cent	0.3	0.0	0.3	0.4	0.3
16	Nitrogen (by difference)	Per cent	85.2	86.0	87.4	88.7	87.9
17	Excess air	Per cent	47	29	9	3	2
18	Temperature of flue gases leaving boiler	Deg. F.	230	295	346	378	434
19	Temperature of air entering below burner	Deg. F.	85	79	92	91	87
	RATES						
20	Heat transferred to water per hr.	1000 B.t.u.	11	34	51	69	88
21	Heat transferred to water per sq. ft. of heating surface per hr.	B.t.u.	550	1730	2650	3580	4520
	HEAT BALANCE						
22	Heat transferred to water (thermal efficiency)	Per cent	44.8	66.4	71.6	72.7	74.5
23	Heat carried away by steam in flue gases	Per cent	10.2	10.5	10.7	10.8	11.1
24	Heat carried away by dry flue gases	Per cent	3.6	4.8	4.6	5.2	5.8

TABLE 3. (CONCLUDED)

Item no.	Particulars of observation	Units	Complete boiler				
25	Loss due to carbon monoxide	Per cent	1.3	...	0.9	1.2	0.8
26	Radiation and convection losses	Per cent	22.0	10.8	8.8	7.6	7.0
27	Undetermined losses	Per cent	18.1	7.5	3.4	2.5	0.8
28	Total of items 22 to 27 or heat in fuel	Per cent	100.0	100.0	100.0	100.0	100.0

sembled boiler as for the previous tests with fire-pot and dome. In all of these tests by adjusting immediately after the Orsat analysis the carbon monoxide content might have been practically eliminated but in so doing there would have been lost the value of conduct of the test under simulative service conditions.

Item 38 of Table 2 showing the increase or decrease in the heat contents of the fuel bed is of general importance, *First*, because it shows the error which may occur in ordinary continuous testing when no means are provided for weighing the contents of the fire-pot, and *Second*, it suggests a means for cutting down the time necessary under the present Low Pressure Boiler Code on small boiler tests without sacrifice in accuracy. It shows that the greatest correction to be applied occurred in trial 65 with anthracite, when the fire-pot contents were found to weigh less by 29 lb. (see Item 16) at the end of the trial than at the beginning, which meant that the heat given up during the trial from the stored up energy in the fire, amounted to 18.3 per cent of the heat of the fuel charged. Had no observations been provided for estimating this depletion of energy of the fuel bed, the efficiencies calculated for this trial would have been about 9 per cent too high. The corrections to be applied in general, however, may be seen to be much less than this and are less for bituminous coal than for coke and anthracite. If trials with coke and anthracite are to be run with continuous method of stopping and starting and no means are provided for weighing the contents of the fire-pot they should, if accurate results are expected, be run for at least the length of time specified by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS Low Pressure Boiler Code.

DISCUSSION

A. J. PURCELL: I would like to ask what sizes of anthracite coal were used in the tests.

H. W. BROOKS: Three-quarters to three inches.

C. H. FLINK: Why the great difference with anthracite coal from 10 to 15 per cent and the gas from 1 to 20 per cent?

H. W. BROOKS: I don't know that I can answer that question except for the fact that with anthracite an appreciable volume of the fire pot was taken up with the fuel itself. With the natural gas burners there was a greater combustion volume and I imagine that the difference could probably be attributable to that. In other words, we have really in the last series of tests an excess of combustion volume over the optimum necessary for natural gas burners. The order of combustion volume required would generally be least for natural gas, next larger for anthracite, still larger for coke, and we know that it should be at least fifty per cent larger and probably 100 per cent larger for bituminous coal; a thing which I think is of utmost importance.

No. 725

DESIGNING AND PLANNING FOR HOME HEATING ECONOMIES

By D. KNICKERBACKER BOYD, PHILADELPHIA, PA.

MEMBER

AS AN architect and consultant, it is a great pleasure and privilege to me to be cooperating with the anthracite industry in its program of information and conservation with respect to the utilization of coal, particularly the size known as buckwheat in residences.

The special object of this paper is to make plain to the public through our membership of engineers, architects and other professions, the need of conservation of one of the most important of Nature's bounties, fuel; one upon which the very existence of mankind depends. While vast quantities of coal still remain in the earth, yet the supply that may be mined with economy or at a reasonable cost, is by no means inexhaustible.

It must also be kept in mind that the cost of each ton of coal, for mining and placing all sizes, either in storage at the mines or in cars for transportation is the same whether it be the coarser coals—stove, egg, or nut—the sizes with which the public is most familiar, or those which have heretofore been called *steam sizes* comprising about 30 per cent of all the anthracite mined.

These small sizes of coal, known by the name of pea and buckwheat, the public must be brought to realize are nothing more nor less than the chips broken from the larger lumps of coal in the processes of mining, screening and handling. Also that unless marketed, they become a dead loss or *overhead expense* as it would be termed by a manufacturer or dealer in any other commodity of commerce, which must be carried by the larger sizes.

Now these small sizes have essentially the same heat value as the larger sizes of anthracite, and the public must be kept more fully informed of the fact and also that by the installation and utilization of the boilers, furnaces or devices especially designed to burn these small sizes, large savings may be made by the householder in his item of fuel expense.

To the architectural and engineering professions this program is of deep interest as it is to all other good citizens. But with the architect, the engineer, the builder, the real estate operator and all others who serve the public through the building construction industry, there is a responsibility and an obligation which augments

Presented at the Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Boston, January, 1925.

the interest which we should naturally have in our service to the public through the construction industry.

Just a word here about the aims and purpose of this campaign of information and conservation. The matter of burning buckwheat coal, as is the case in almost any economic problem, is very simple when once the facts are fully understood. Many of these facts about the utilization of anthracite are apparently known, but have not frequently enough been promulgated or been given detailed consideration or put into actual effect with the designing, planning and specifying of the home.

Take, for example, the case of the average architect. I am not speaking now of the larger offices which retain technical advisors and engineers for all of their work or of those architects who, through long experience, may be competent to handle residential heating problems. I am referring to the average practitioner and especially to all younger architects and those starting their careers without a wide background of experience. The architect must not only be familiar with all forms of design but he is expected to be conversant with the best methods of construction, and to be familiar with building and heating codes and sanitary requirements and to know all about most materials and equipment from a common brick to a complicated elevator.

All this knowledge of design and technical details naturally cannot be learned from books or schools or quickly picked up from practical experience or through private investigations. The architect must perforce profit from reliable information made available to him from all sources. The Bureau of Standards, the *American Society for Testing Materials*, the *American Institute of Architects*, various engineering societies, including our own, and many other such organizations as well as modern industrial associations, are constantly working to produce additional knowledge for the professions, the building industry and the public. Only by taking cognizance of all this information can the practicing architect—or any one of us—render the proper service to his client.

Individually an architect may be well versed in heating but collectively the profession is dependent on others allied with it for information. There are many technically trained heating and combustion engineers throughout the country whose life work is the study of heating problems. They, too, look to this Society and to contractors and manufacturers cooperating with it for cooperative investigation and collective information. It is to take part in such work and to augment information about the burning of buckwheat and to make it available to architects, with whom the plans for buildings originate—and to builders who construct without architects, and to all others who should be interested, that this campaign of the anthracite operators was inaugurated.

It should be the function of the anthracite industry and of the retailers who distribute its products, cooperating with appliance manufacturers and heating engineers, to keep architects, builders and the public fully informed as to the products of the coal mines and apparatus for their utilization through proper combustion. Happily this program is now well under way and it is my hope to see the fullest cooperation between the anthracite operators, this Society, and all of those concerned with the construction and maintenance of buildings.

With this introduction, I want to feel, therefore, that I am speaking on behalf of all those who plan, specify, design, construct, install or sell anything connected with heating the buildings in which people live and work. We must all do these

various things in such a way that the comfort and convenience of the householders and all occupants shall be met with the truest economy to themselves and to all industries upon which they rely for shelter—including that prime essential to life itself in cold climates—heat.

With respect to comforts and economies to be derived through heating, it is the obligation of all of us to afford every possibility for the proper utilization of all the sizes of coal which are mined. It should be the function of the architect and the builder to utilize properly all good products of the quarries, mines, forests and factories. This applies to the fuel burning equipment as well as to the construction of buildings.

For instance, in the matter of a common building material like brick some of us know that when clay is burned in kilns, the limitations of manufacture are such that no matter how carefully placed or burned those brick farthest from the heat will be the least burnt and in consequence softer than the others. In some localities these brick are known as salmon. While they are from the same clay as the others, they are not as durable, when exposed, as the better burned brick but there is always a place where they may be properly and advantageously used. Intelligent and thoughtful architects and builders make proper use of them in construction or the cost of the other bricks would be greatly increased. Building code officials and fire insurance organizations cooperate in arranging for their proper utilization also in allowing 8 in. brick walls for small dwellings instead of 12 in. walls as heretofore required. Such provisions as here stated make for the ultimate economy of the consumer—who usually is not brought face to face with these features as he or she is with coal.

In the latter case they add to their own costs—household expense—by their own choice. And let us emphasize this fact once again by repetition; "the householder either *increases* or *reduces* a very essential item of living expenses by his or her choice of fuel." There is no getting away from that.

Take the case of lumber. There used to be a time when not only architects and builders but the uninformed public indiscriminately demanded the use of clear-wood—"without knots, blemishes or other defects"—as the specifications used to read. When our forests could not grow fast enough to furnish all wood required for construction to say nothing of only selected clear gradings, it was soon realized that all of a tree and not only its heart would have to be used. Now, appropriate and perfectly proper uses are being found for the various grades—and the cry is still more wood.

It is the same with tiles, another building product with which all are familiar and the product of another industry to which I act as consulting architect. It was long the custom to call for all tiles to be of the "very best quality, straight, even, smooth, true, etc., etc." without regard to the kind or cost of the house or knowledge of how tiles were made and graded. Years ago I was guilty, myself, of thinking that tile manufacturers deliberately made a first quality, a second quality and a third. I soon learned that there is but one quality made, for the manufacturers take the same care in preparing the clays, in shaping the units and in placing tiles in kilns—just as with brick. But when the tiles are removed from the kilns it is found that through forces beyond the control of man some are straighter than others, in some the glaze is fuller than others and so on. Thus the manufacturers feel obliged to grade the tiles and instead of 1st, 2nd and 3rd grades, we have Selected, Standards and Commercials. Informed architects now specify Commercials for moderate priced dwellings whose owners get the same quality

as the moneyed man with his Selected—but the effect, while not the same is perfectly satisfactory for the intended use and at the lower cost.

So it is with glass and other manufactured products and especially with slate, marble, sand and other natural products used in buildings. But I will not go into further details.

These brief references to materials of construction will serve to illustrate the point I have in mind about utilizing all the products of the coal mine where the chief difference is size available for domestic uses. With respect to coal as a vitally important material for storage and use in the home after it is erected—architects and builders should have the same knowledge and forethought as to the materials of construction.

They should know that all sizes of coal must be utilized to balance the products of the mines just as they must balance the products of the kilns, the forests, the factories and the quarries.

The public too must do its share. It has long ago learned economies with respect to some other products. Take beef for instance. We all know, especially the ladies, that not all of a carcass can be porterhouse and tenderloin steaks—there must be sirloin, rump and other cuts. All are nutritious and make good eating—but the prices are different and there are also different ways of cooking them.

With respect to burning *any* coal economically—more especially the finer "steam sizes" of anthracite—both the architect, the builder and the householder should share the responsibility for the proper construction of dwellings—in that unnecessary leakage of air be obviated, but more especially should they look to the proper design and arrangement of chimney flues that proper draft be obtainable, and upon which so greatly depends the satisfactory and economical utilization of any fuel. And, if we are to aid in the conservation of anthracite, if we are to make an earnest effort to "reduce our coal bills" we certainly cannot overlook the proper planning of that prime necessity—the chimney—more important in coal conservation and household expense reduction than any other practical feature of house planning. This is not an exaggerated statement. It is a plain, unvarnished statement of fact to which any experienced engineer will subscribe.

Now let us hope that through cordial cooperation between architects, builders, all elements of the construction industry, home owners, heating appliance manufacturers, coal operators and also coal dealers, there will be evidenced from now on a greater knowledge of the simple requirements of proper combustion which will be reflected in better designed and constructed dwellings—more especially the chimneys upon which depends the intelligent utilization of anthracite coal of all sizes, including the equally valuable lower cost buckwheat.

Speaking of architects and of such builders as construct and sell dwellings without architects, the responsibility begins with these six things.

First—The size and height of flues for all heating and cooking apparatus.
Second—The location and construction of all chimneys.
Third—The specifying and installing of proper heating and cooking apparatus.
Fourth—The locating of coal bins.
Fifth—Providing facilities for the removal of ashes.
Sixth—Making walls, roofs and openings tight.

Briefly referring to details of these features:

Flues

The commonest errors (and they are, unfortunately, far too common) are that

flues are built too small and that not enough of them are provided. It should be axiomatic that every flue should be amply large and high for the draft necessary to burn the required fuel in whatever apparatus may be attached to it and that each and every appliance should have a proper size flue of its own.

An exception might be made in the case of a laundry stove connected to a range flue if used but one or two days a week. It only costs a few cents more for the next larger size of flue lining in a free standing chimney and not one cent more when part of a wall—so why skimp in size or number of flues?

That flues have not always been of the proper size cannot be blamed on the architect or engineer or even the builder unless influenced by false economy, as knowledge concerning requirements has heretofore been lacking. It is my privilege to be a member of the Sub-Committee VIII on Design of Chimneys and Flues of the Code Committee of this American Society of Heating and Ventilating Engineers. This Committee has at last developed what it considers to be accurate data from the standpoint of engineering in regard to sizes and height of flues—which should, of course, be lined with fire clay lining. This data, pending the final adoption of the Code itself may be found in The Guide and also in *Flues and Flue Linings*, just published by the *Eastern Clay Products Association.* In the latter publication, it is recommended to insure the tightness of each flue, that a smoke test be made on each flue by the contractor erecting the chimney and that each flue must be made absolutely tight, if not so, before scaffold is removed or chimney accepted.

Chimneys

In addition to being safe, sound and tight—and too many are not tight enough to give proper draught, hence the above requirement—the location of chimneys should be carefully studied. Too often they are located not where best service requires them but so that they will not interfere with placing stairways, halls and all other features or so that they will occur in corners or closets. Such considerations frequently influence location without due regard to the points of the compass, the proper location of the apparatus or its accessibility with respect to the coal bin and the removal of ashes. Why should we not all give more thought than has heretofore been given to these very practical aspects? This is another opportunity we all have to cooperate in making for further household economies.

Why do so many architects and builders in various localities apparently abhor a chimney that projects high enough above the roof to give adequate draft? We in America seem to have a prejudice against what in England and other European countries can be made, and are made, features of no mean importance in the design and architectural effectiveness of buildings. If architects, encouraged by engineers, make larger and higher chimneys the fashion, they will be followed everywhere.

Now to emphasize once again the prime necessity of cutting down fuel bills with no added inconvenience. Only by reiteration—by the oft repeated statement of simple facts well known to a few—will the truths of the best way to burn low priced, not low grade fuels, be brought home to the public.

Keep in mind that the buckwheat and all the other "steam sizes" of anthracite cost just as much to mine and distribute as the coarser sizes with which householders are most familiar; also that 30 per cent of *all* the anthracite mined is to be found in the small sizes and must be disposed of—burned in heating and other furnaces, else the remaining 70 per cent of the output *must* bear the entire cost of production, and must of necessity be sold at a higher price.

Here I will again suggest the analogy of the family insisting on porterhouse steak in preference to sirloin cuts that sell in the markets at a much lower price per pound. Most of us are insisting on the porterhouse steak instead of sirloin when it comes to anthracite as a parallel and are still paying for the choice cuts of anthracite while the small sizes of the same basic material are accumulating at the mines in vast quantities.

After giving home owners ample flues and tight chimneys well located and having the cooperation of coal operators and coal dealers on the one side and of builders and home owners on the other, there is still an obligation on the part of architects and engineers and on the part of manufacturers of heating and cooking appliances and of heating contractors, to specify, make and install appliances in which buckwheat coal can successfully, conveniently and economically be burned.

So far as we of the architectural and building fraternity are concerned, the matter of buckwheat burning easily divides itself into two major classifications. One is equipment in new buildings; the other is equipment in existing buildings.

For new buildings we may specify and have installed either new apparatus of the magazine feed type, or we may use any present type of apparatus and install with it smaller sized grates for burning buckwheat and a system of mechanical draft—unless the chimney is high enough or the building so located that natural draft will be sufficient.

In existing buildings, too, the magazine feed type of apparatus can be used when the flue is large, tight and of sufficient height.

But where favorable chimneys are not encountered or where the condition of the existing heater or boiler would not seem to warrant the expense of its replacement by a new one, auxiliary appliances may be installed in the existing apparatus by means of which buckwheat may be burned efficiently.

I will not go into further details concerning apparatus or appliances as these features will be cared for in the papers of others and will, I trust, be fully taken up in the discussion which it is hoped will follow.

And now a few words about the lowly but, in my opinion, important subject of:

Coal Bins

There, in the cellar or basement, frequently tucked away in the darkest corner—farthest removed from the heater, the stairs, the daylight and the artificial light and almost inaccessible to the owner, the servant and the coal dealer, alike—is the object designated as the coal bin.

In the specifications of thoughtful and informed architects these utilitarian features are given adequate consideration and careful mention. In others I can imagine somewhat as a postscript at the end of *Carpentry,* words to this effect—"Properly and securely construct two coal bins where shown or directed, sides to be 4ft. 6 in. high and each to have one loose board section with shovel hole."

Such a specification might perhaps just as well read like this, "Before leaving the job the builder shall not forget to assemble all boards remaining which have not become totally unfit through use as scaffolding, and shall find some spot in the cellar not occupied by water and gas meters, soil pipe clean-outs, valves, shut-offs, floor drains, electric conduits or other impediments and shall construct with these boards and any new ones found necessary such sized coal bins as will require the least amount of material, will reach from the corner farthest removed from the

street or driveway to half way across the nearest window and be of size sufficient to hold about one-third of a season's supply of coal."

Between us I hopefully look for the day when the great importance of properly locating and constructing coal bins will be given full recognition by all concerned.

Such coal bins will doubtless be of masonry from floor to ceiling, with tight walls and ceilings. They will be located within but a few paces of the apparatus and in a position where the dealer can, through a specially built opening, chute the buckwheat or other sized coal in from the street curb or the driveway. In order to reach them it will not be necessary to grope one's way, ducking heat pipes, during a long unlighted walk from the apparatus. Thought will be given to every convenience and to every economy.

And this is where further cooperation will come in—especially between architects, engineers, builders and coal dealers. Waste of labor and of money must be eliminated. Think of the additional cost to the people of any community in having coal carried that might be chuted! In talking with a prominent official of a Coal Merchants Association, he showed me but one instance of this waste. A coal bin properly located in a house using 24 tons of coal had the one available window placed on the side of the building instead of the front—where it might just as well have been. Because of this fact the dealer was obliged to have the coal carried instead of chuted. The owner had to pay 50 cents more for each ton thus carried. In other words the owner paid an annual tax of $12.00 or the equivalent of $1.00 a month because of the way his cellar was arranged. And of course, with each bill he objected to the charge, upon which the dealer probably made no profit and certainly suffered the added annoyance of a dissatisfied customer.

Let us look at this waste in the aggregate. In that community one million and a half tons of coal are burned annually in residences alone. This official estimated that fully 800,000 of these tons are carried at an average cost of 50 cents each. In other words, if all of that coal could have been chuted (of course, not *all* of it could be and in some cases the difference would be justified) it would save the people in that one community $400,000 a year. A great part of this sum therefore is chargeable solely to bad planning and could be saved.

Let us speed up this cooperation and start now to plan and build correctly while installing apparatus to burn buckwheat that will cut down coal costs, conserve materials and make for greater comfort and economy all around.

And now a word or so about combustion and the efficient burning of buckwheat. We will assume that ample provision has been made in the construction of the dwelling for a chimney suitable both in area and height; that the coal bins are of proper size and in reasonable proximity to the furnace, and that the occupant of the house has made up his or her mind that household expenses are to be reduced in part at least through the coal burned. It is also assumed that the furnace in which the fuel is to be burned is of ample capacity to heat the house. Now Pennsylvania anthracite coal is universally recognized to be the solid fuel best suited for domestic use when cleanliness and convenience are considered, and the least expensive also when the finer sizes are burned.

Buckwheat coal may be burned practically in any heating boiler or hot air furnace where the chimney draft is sufficient either in itself or is augmented artificially.

The same heating results are thereby obtained as would be if the stove or egg sizes were burned, costing about 100 per cent a ton more. In such instances, the motor driving the fan would be under thermostatic control, and in operation only when the

draft proved insufficient to keep the house warm. Such devices require the minimum of attention and are in every way dependable. Special grates will be required with this fine coal, but are easily obtainable.

Removal of Ashes

Much of what I have said about facilities for receiving and storing coal applies to the storage and removal of ashes.

Too little thought is also given this subject which touches the convenience of all householders. The methods of care and removal will depend upon whether the residence is in city or country and if in a city whether or not there is periodic municipal collection.

In any event, without at this time going into details, one essential should be a requisite in every home. That is an ample cellarway—whether of the sloping-door type or the stair and area type. A sufficient number of metal containers should always be provided and the necessity never arise for loose storage on floors or for transferring ashes to wooden non-fireproof boxes for removal through narrow high slits of windows in front of which so often there are lilies and other tall plants blooming when spring comes round again.

Tightness of Construction and Other Economies

For these many suggestions could be offered but it is beyond the scope of this paper to describe them in detail. I refer to walls which should be substantially built everywhere and furred in most localities as a protection against condensation and radiated cold; to fire-stopping of masonry, as well as of frame, walls—which also acts as draft-stopping; to window and door frames which should be standardized with lugs and then calked so as to doubly prevent air leakage; to proper weather stripping as a protection against infiltration; to cellars that are ceiled or floorings that are doubled over cellars, to roofs that are insulated and to pipes in cellars and outside walls that are covered.

All these features together, if incorporated in every home, would add but a small percentage to the total cost, would return annual savings, increase comfort and conserve coal. It is time that the average home owner came to a realization of the high cost of cheapness as compared with building and equipping well in the first place, having low maintenance charges and living happily ever after.

No. 726

SMALL SIZE ANTHRACITE FUEL AND EQUIPMENT FOR ITS ECONOMICAL UTILIZATION

By C. A. Connell,[1] Philadelphia, Pa.

NON-MEMBER

THE problem which confronts the anthracite industry today is one of economic nature, to be solved only by the complete utilization by satisfied consumers of all the combustible products of the mines. As it has always been the privilege, as well as the duty of the engineer, to conserve the natural resources of the country, it is especially gratifying to be able to present some facts on coal and its economical use.

Looking retrospectively into the history of anthracite, it will be recalled that ever since hard coal was first used, the larger sizes have borne the brunt of the production cost of both the larger and the smaller sizes. This was due, in the beginning, to the belief that only larger sizes could be burned to generate steam; the smaller sizes being deemed a necessary, but useless by-product. Further and more intensive study of combustion has developed the idea that the smaller sizes could also be burned efficiently, when given the proper mechanical auxiliaries to support the coal and to supply the draft. It is natural that this work was started by the engineers at the mines and the engineers of the large individual power plants, since they alone could afford the expense and had available the engineers capable to do the research which was required. Following the introduction of the smaller and less expensive coal to the power plant came the advent of the central generating plant, and with it the conversion of individual high-pressure units to low-pressure heating boilers. It was therefore only natural that the experience in the high-pressure plants should have been used to advantage by modifying the equipment for burning this coal in high-pressure boilers to adapt such equipment to the heating boiler. It is also natural that the heating and ventilating engineers throughout the anthracite territory should have gone another step further in the same direction, and they brought out systems for efficiently burning the small coal in the home. So that today there is equipment on the market by means of which the low priced sizes such as buckwheat coal may be burned satisfactorily in the smallest home, as well as in the largest power plant.

Even this fact, however, has not caused the general acceptance of the small coal by the public, and it is still a fact that the larger sizes must even today bear more than their proportionate share of mining costs, and causes what the public often

Presented at the Annual Meeting of the American Society of Heating and Ventilating Engineers, Boston, January, 1925.

believes to be, an excessive price to them on the larger and so-called domestic sizes of fuel. Any reduction in cost of this fuel cannot be accomplished until the above result has been accomplished, and since this result can only be accomplished through a program of educating the public in the correct uses of the smaller sizes, the anthracite industry has recently established service exchanges in six of the principal cities of the East. Competent engineers are in charge of these service stations, and advice may be obtained from them without cost which will enable the public to reduce their annual fuel bills, while still taking advantage of the same anthracite fuel which is so well established as the most satisfactory and safest heat producer. If, as is often the case, it is impossible for the engineer in charge of the service station to solve the problem of the person who is inquiring without first making an examination of his plant, other engineers are available who will be sent out to make such an examination, and to report on the economies which may be accomplished. In other words, the anthracite industry has now adopted the same method of service which has proved to be so necessary by all the successful public utilities companies, and are striving to be of real service and aid to the public. In addition to the permanent exhibits which are mentioned above, and which are located in New York, Brooklyn, Boston, Philadelphia and Washington, a travelling exhibit, spending one week in each of the 20 medium sized cities throughout the anthracite territory, is giving the same sort of service to our consumers in those cities.

While the advice of the engineers is given on burning both the larger and smaller sizes, it is particularly concerned with the latter division, since it is essential that if small coal is to be burned efficiently, it must be consumed in equipment especially designed for this purpose. A laboratory has recently been established, under the direction of Dr. Fernald, head of the Towne Scientific School of the University of Pennsylvania, where equipment for burning small size anthracite will be carefully tested, and unbiased reports prepared. In this manner, knowledge of equipment which fulfills the necessary requirements will be obtained and its satisfactory performance will be approved. Likewise data on devices which will not efficiently perform will be in our possession and this information will act as a protection to the public, and to you as their consultants, against the impositions of inferior and unsound devices.

Equipment already tested and found satisfactory is being exhibited at the various service exchanges, and they naturally divide themselves into two distinct classifications: The first, where a new building is being erected, or where a boiler is in such condition that replacement is necessary; and the second, for installations in which the boiler is in such a condition that replacement would not seem to be warranted.

To burn buckwheat in new buildings, we recommend and exhibit the magazine type of heater. This heater provides for obtaining the requisite draft without mechanical aid by the maintenance of a relatively thin fuel bed, which reduces the resistance to the flow of air through the fuel. In order to maintain the thin fuel bed, and at the same time to eliminate the necessity for frequent firings, the magazine feature is included, so that the coal is fed to the grate automatically, as coal is consumed. By the installation of such a heater, a double economy is secured in that not only is a cheaper fuel burned, but also the time of the person tending the boiler is conserved, as it is only necessary to fill the hopper once a day and shake the grate once every eight hours.

However, in all installations where small coal is being burned, primary consideration must be given to the draft, and the manufacturers of magazine feed boilers agree that whenever consideration is being given to the installation of their heaters,

they should first be consulted, so that they may make certain that the flue and stack conditions are such as to insure against insufficient draft, which has caused a great deal of trouble in the past, by the installation of these heaters through uninformed heating contractors in buildings where chimney conditions were such as to make satisfactory performance impossible. Such an occurrence will be easily eliminated by pre-inspection, as above.

For boilers which are already installed, and whose life is good for at least four or five years, auxiliary equipment which may be installed in these boilers has been perfected, and in the large heating plants conversion to small coal may be effected through the installation of a specially designed grate, such as the Pyramid, the Herrington, the Pin Hole—either water cool or air cool—to be used in conjunction with forced draft, and preferably with automatic control. Such equipment may be installed in practically any heater or furnace with the exception of the warm-air furnace, which is generally unsuited for forced draft, as the gases volatilized from the coal by forced draft are generally at a pressure slightly above the pressure in the leaders and users, and any leaks between the castings cause a flow of gases from the combustion chamber into the ducts through which the heated air is distributed to the various rooms of the house.

The equipment referred to is for installation in fairly large boilers and it is unfortunate that exactly the same principle cannot easily be adapted to the small boiler in the home. Usually, the home is furnished with a small vertical boiler having a diameter depending on the size of the house, ranging from 16 in. to about 31 in. In order not to make necessary frequent firings in such a heater, a thick fuel bed is carried, and it is not uncommon to fire 6 in. to 8 in. of green coal at one time. Under these conditions, when firing with forced draft, gases are generated at a rate which is faster than is the ability of the stack to remove them, and causes a congestion of gases just above the fuel bed. This results in leakage of these gases through the opening in the boiler to the basement, and then throughout the house. This is, of course, very annoying, and furthermore, as the gases which are volatilized are only partially consumed and contain a high percentage of carbon monoxide, they are subjected to instantaneous combustion, when under slight pressure the gases are ignited by a flame leaping through the fuel bed. Explosions may result of more or less consequence. Such explosions may be minimized and even eliminated by careful and correct firing, which provides for leaving red glowing coals in evidence on top of the fuel bed after firing.

To care for this condition, and to enable the home owner to satisfactorily burn buckwheat in his existing heater, he can use an induced-draft system, this system being merely a modification of that with which you are all familiar in its application to the larger power plant; the fan being inserted in a by-pass of the smoke pipe from the boiler to the stack. The fan operates automatically to maintain a predetermined temperature, and merely acts as a booster to the stack. The by-pass is furnished with a balance-damper which provides for maximum utilization of and operation on natural draft when the desired temperature is reached. In addition to the induced-draft fan and automatic control, the system is made complete by a shaking grate designed with $^3/_{16}$ in. apertures for supporting the buckwheat coal. With this installation a slight negative pressure is maintained above the fuel bed at all times and leakage of gas made impossible.

Time will not allow a detailed discourse on all the various equipment, it being desired at this time to merely place the idea of burning small anthracite before you, for your consideration, and to emphasize the fact that equipment for burning this coal is obtainable at prices commensurate with the savings effected, and by means

of which the public may be enabled to materially reduce their fuel bills while still enjoying the luxuries of anthracite.

Summarizing, it is my direct purpose in coming before you today to enter a plea on behalf of the anthracite operators, to enlist your hearty cooperation in this comprehensive service-to-the-public program which can only react to afford you as consultants to and advisor of the public, maximum benefit.

JOINT DISCUSSION

PRESIDENT ADDAMS: Both these interesting papers deal with a problem that probably affects 25 million of our eastern population and I hope for the broadest and most beneficial discussion of this question.

QUESTION: What draft is required for domestic heating?

C. A. CONNELL: We have found when using a forced draft in a domestic heater that there is required approximately $^1/_4$ in. on the water gauge, in the ash pit; when using an induced draft fan, the average domestic installation requires approximately $^2/_{10}$ in. on the gauge inserted just above the fuel bed. I am referring there entirely to buckwheat coal.

ALFRED KELLOGG: I was unfortunately unable to hear much of Mr. Boyd's paper. I would like to ask Mr. Boyd if he touched upon the use of the finer sizes of anthracite mixed with the coarser sizes, such as egg.

D. KNICKERBACKER BOYD: I have attended a meeting or two where the question has come up as to what is meant by "mixing," which in my opinion should be called "layering." I would like to suggest, however, that Mr. Connell as a combustion engineer respond to that question. I have a friend in Philadelphia who uses the buckwheat to bank his fire at night and says it isn't a question of economy with him alone; it is a matter of convenience. He would use buckwheat for that purpose if he had to pay the same price as for the other coal.

ALFRED KELLOGG: It seems to me that we have been endeavoring to classify methods of burning buckwheat coal, or rather No. 1 buckwheat coal, in two ways: In the new installation where a furnace is built for the purpose installed, and the other is the alteration of older installations and the use of forced drafts. It seems to me that the use of forced drafts or induced drafts for domestic installations is only a short lived proposition.

In a domestic installation motor and blower are naturally located down on the floor near the furnace or boiler, in the dirt and ashes, all of which tends to wear out the bearings. It seems to me that two or three years' life is what the average man is likely to get out of such an installation and it hardly warrants the expenditure.

I believe that fully 75 per cent of the household installations already made, where provision has not been made for boilers specifically designed for No. 1 buckwheat, present difficulties that can be readily met by a mixture of the coarser sizes of anthracite with the buckwheat. If we can use a half and half mixture of egg coal with No. 1 buckwheat whether it is layered or otherwise, we will cut the cost of our total year's expense of coal at least 25 per cent.

H. W. BROOKS: In answer to the point which Mr. Kellogg has brought up, it may interest you to know that this question has been dealt with in the last report of the U. S. Coal Commission to President Coolidge. It was suggested in that re-

port that the Bureau of Mines report on the additional draft necessary to force air through mixed sizes of anthracite. Later on the *Anthracite Coal Operators' Association* approached the Bureau with a specific request for such tests. About a year ago tests were commenced.

Work was divided up into two classes: First, the flow of air through unfired aggregates and, Second, the flow of air through fired aggregates. For unfired aggregates alone there are seventeen conditions which have to be investigated and correlated to get translatable results. To date, laws for nine of these relationships have been deduced. At the request of the *Anthracite Coal Operators' Association* and the U. S. Coal Commission a progress report describing this work, will be issued shortly.

President Addams: If I may speak of my own experience, when I bought the house I live in it required 40 tons of coal, which was appalling to me. I started out by taking the lowest priced large anthracite, the egg, and the lowest priced small anthracite, the buckwheat, and instead of 40 tons I put in 15 tons of egg and 15 tons of buckwheat and that sufficed for the year. The space in the grate was too large to admit burning buckwheat coal alone, so the next year I put in 8 tons of buckwheat and 8 tons of egg and changed the grate to one having about a $^3/_8$ in. air space, a little larger than I think just right for buckwheat coal. I could, however, run along in the spring and in the fall very satisfactorily with buckwheat alone and then during the real heating weather I fired egg coal and top dressed it with buckwheat.

The coal consumption was less because I was able to control the free burning of the large coal very much better spring and fall.

I do feel that buckwheat coal is the salvation of the people who live in that anthracite region where we look on these people who sell anthracite as having hoofs and claws and coming down with red limousines, owning their own railroads and being real vicious, money-accumulating barons, who own the mountains of our State.

C. A. Connell: The remarks of Mr. Kellogg and Mr. Brooks are very interesting to me. We have gone into the mixing of the different sizes of coal to some extent and also into firing in layers. The Philadelphia & Reading Co. attempted through their engineers in Pottsville to develop the burning of buckwheat mixed with various sizes of the larger coal. They finally gave that up as a bad proposition. Their reports showed that when you mix the two coals, first of all you raise the amount of clinker and second that the smaller coal will burn out more rapidly than will the larger coal and the ash which is formed by the smaller coal burning out collects around the larger, resulting in the carbon content in your ash increasing.

As to the effect of dust on the motor, that is of course a factor; however, I take exception to Mr. Kellogg's statement of two years. They have installed in the Harrisburg district over 5000 forced draft apparatus for burning river coal. The great majority of these have been installed for four and five years and were giving complete satisfaction.

E. H. Lockwood: For nearly twenty years my ten-room house has been satisfactorily heated by No. 1 buckwheat coal, or its equivalent in size, yard pea. This has been accomplished by supplementing the buckwheat by a small amount of gas coke, which removes the chief drawback in burning fine coal. The object of adding coke is to get a hot fire quickly in the morning, or at other times when extra heat is needed. The usual proportion of coke to coal was about one to five.

Coke was chosen for a supplementary fuel because of its quick burning characteristics, and also because a supply was always available during the winter at a fixed price. Wood has been used occasionally instead of coke, and serves the same purpose although a little more awkward to handle. My experience demonstrates that buckwheat coal will give out a considerable amount of heat once the fire is hot, but help is often needed in getting the fire into a hot condition.

Smaller charges of buckwheat must be used, hence more frequent firings are needed, say morning, noon and night in cold weather. More care in firing is required, such as covering half the fuel bed and waiting for the fresh fuel to kindle before adding the rest. Buckwheat coal is therefore more troublesome to fire especially in cold weather, but this is partly offset by the advantages of the small fuel in mild weather. Since the use of buckwheat coal will save about one-half the annual coal bill at present prices, a little extra trouble should be expected in return for the saving.

No change of grates is required in burning buckwheat coal, owing to the characteristic of the small particles in fusing together as they burn. By careful shaking the fine dust can be removed from the fire pot, leaving a coarse ash above the grates. The best time to shake out the ash is when the fire is hot, since the burning fuel is then most firmly united, hence less liable to fall through the grates. Best results will be obtained by carrying the fuel bed at level of the charging door, shaking out ash as required to make room for more fuel. The draft required in burning buckwheat, supplemented by coke or wood as described, is not excessive, and probably not greater than can be furnished by the average chimney. Forced draft removes the necessity of relying on coke or wood to start up the fire, but is attended by the extra cost and complication of the blower, which is not usually justified in a small installation.

C. K. DENNIS: What is the equipment in your house?

PRESIDENT ADDAMS: An ordinary boiler; the grate is a rocking grate with bars about six inches wide.

J. F. MCINTIRE: I am surprised to hear the suggestion of mixing buckwheat and the larger sizes of anthracite. After looking into my coal bin, where I have a supply of stove size, I thought the operators had gone just as far in that line as could be done.

I forgot to ask the speaker the average difference in ash content or B.t.u. content of stove size and the No. 1 buckwheat as it comes from the mines.

C. A. CONNELL: Taking up the questions in their order, the difference in B.t.u. content differs with the various mines. However, I believe that it is safe to say between 400 and 500 B.t.u.'s is the difference in heat content between nut and buckwheat.

ALFRED KELLOGG: Prof. Lockwood is particularly well situated in burning this fuel, from the fact that he is using a hot water boiler. I should question whether the same results would be attained in a boiler on a so-called vapor system.

Mr. Connell brought up a question which, it seems to me, is particularly pertinent. He describes the tests made at Pottsville. These I assume were made on anthracite coals mined in the southern district, known as red ash. The characteristics of this coal are entirely different from the anthracite mined in the northern district. I should expect the clinkering that he describes, and yet you take the white ash coal or northern anthracite, it does not give the reaction that he describes. We

get a lot of bone coal especially in the egg sizes. In burning that in my furnace I find a lot of the bone will come out through the grate and leave what you might call clinker. It isn't exactly that but a lot of it is bone. If I mix buckwheat with that, I find all that bone goes into white ash. I believe it is because of the surrounding of those bone particles by the finer anthracite which produces a higher temperature at that point and disintegrates the bone particles to the point where we get no clinker whatever. I don't believe that is possible with red ash coal.

H. F. HUTZEL: We have run many tests on buckwheat coal and I have burned buckwheat at my home. It has been our experience that it is practically impossible to get satisfactory results with buckwheat coal on an ordinary shaking type of grate. We made many analyses of buckwheat coal and have never in our experience found any coal with less than 20 per cent ash. We know, too, that the ash in buckwheat coal slags. You don't get a free ash as with other sizes of anthracite.

With a shaking grate, in order to clear your grate, you create a crack in the slag which permits a large percentage of the buckwheat coal to run through, and if you take an analysis of the refuse in the ash pit, you will find that it contains quite a large percentage of carbon. With the revolving grate you grind off this slag at the bottom.

Furthermore, the voids in the buckwheat coal are very small; you get the greatest void around the perimeter of the fire pot. You will find, too, if your fire box is full in the morning, after burning four or five hours, the coal burns out around the edge. You can't satisfactorily use a poking bar in there to poke that ash down and it is very desirable if you are burning buckwheat coal to have each individual grate bar revolve separately to permit you to clean the grate along the edges. I don't believe that you can ever educate the people to exercise the patience that is required to use that type of fuel. A few men like Prof. Lockwood and a few others here who are interested in combustion or who are desirous of getting good results have the patience and they will exercise it, but the average consumer doesn't bother about that. He will shake his grate the same irrespective of whether he is burning buckwheat or large sized coal. The result will be that he will lose a large percentage of his coal through the ash pit.

A. J. PURCELL: I would like to ask what he finds to be the necessary draft through buckwheat coal in a magazine type of boiler.

C. A. CONNELL: I personally have conducted very little work on the magazine type of boiler. I have run it on the same stack required to operate the average boiler under the large coal and know that it will operate successfully at about fifteen hundredths of an inch. I see a number of the representatives of magazine type boilers and they possibly can give you exact data.

W. H. WILSON: Prof. Lockwood ran a recent test on that. Do you recall?

E. H. LOCKWOOD: I think about twelve or thirteen hundredths.

H. M. HART: I want to ask if that was the draft over the fire?

E. H. LOCKWOOD: That is the draft in the smoke pipe.

H. M. HART: What would the draft over the fire be?

E. H. LOCKWOOD: It would be practically the same.

PRESIDENT ADDAMS: Both questions are in order. I think probably Mr. Hart would like to have his question answered.

H. M. Hart: Perhaps I am confusing the question. The question, was, what is the draft necessary in the magazine type of boiler, but the question I would like to ask is, what draft would be necessary over the fire to burn buckwheat coal compared to burning stove sized anthracite.

C. A. Connell: In answer to that I will say that it would require the same amount of draft. The reason is a thinner fuel bed is maintained than is the case with the large coal.

J. F. McIntire: The reason why I ask about the ash content and the heat content of No. 1 buckwheat is that I have always found a greater difference than the speaker gives of 400 or 500 B.t.u.'s per pound between No. 1 buckwheat and stove or egg sized coal. It is very important when you consider the difference in price between No. 1 buckwheat and your larger sizes of coal.

C. A. Connell: In replying to Mr. McIntire, the tests which we have made during the last year show the smallest average is around 12,250. The highest was 12,400.

H. W. Brooks: About a year or two ago the Bureau sent two men into this State to take samples of coals from dealers' yards. I don't know how old the coal was that we were sampling at that time, but I remember particularly the highest figure which was shown on several hundred samples was in the neighborhood of 50 per cent non-combustible. That is frankly an exaggeration of average conditions of coal, but I think that the anthracite operators can well give attention to possible deterioration of coal after it leaves their own yards. I don't know whether that is the entire cause of the deterioration or whether it is a let down in the washing processes themselves, but certainly the coal as delivered to the customer's bin does not compare favorably with the coal which we have analyzed in the various bureau bulletins which we have put out on anthracite in the past.

C. A. Connell: May I ask, Mr. Brooks, if you made any recent tests on the B.t.u. of buckwheat?

H. W. Brooks: We have, and it shows up decidedly better than in the past two or three years.

C. H. Flink: I had an experience with coal lately. We took the regular sample in the approved *A.S.M.E.* way and found they had 21 per cent ash and immediately got in touch with a couple of the largest dealers in Buffalo and asked them what should be the percentage of ash in anthracite coal. They made a couple of guesses and then I told them why I wanted to know. They said they would look it up. I called again the next day and they told me, "Well, we don't buy it that way; we get what they send us from the mine, made up in their regular way."

D. Knickerbacker Boyd: Just before closing I want to refer to this subject of the stack again and the idea of recommending the layering of coal. I have two or three friends who have tried layering without success and it is due probably to the fact that they live in houses with poor chimney draft. In my firm's architectural practice we have always tried to have our flues not only large, but amply large to take care of any condition like this.

No. 727

THE SEMI-ANNUAL MEETING, 1925

ACCEPTANCE of the Code of Minimum Requirements for the Heating and Ventilation of Buildings made the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, held at the Hotel Traymore, Atlantic City, N. J., June 15 to 17, outstanding.

The Meeting was called to order on Monday, June 15, with Pres. S. E. Dibble presiding. After a few preliminary remarks the president called for reports of committees. Perry West presented the report of the Guide Publication Committee and the report from the Committee to confer with the *American Institute of Architects*. These reports were followed by the reports of the Committee to Prepare a Code for the Guidance of the Membership Committee and the Committee on Increase of Membership. The remainder of the first session was given over to professional papers.

The second session was called to order at 1:30 in the afternoon of the same day. Harry H. Harrison, representing Mayor Edward L. Bader, welcomed the members of the Society and presented President Dibble with the key to the city. This was followed by the reading and discussion of professional papers.

Amendments to the Constitution and By-Laws, proposed by the Kansas City Chapter and seconded by the Illinois Chapter and the Pittsburgh Chapter were presented to the Meeting for the consideration of the members.

W. H. Driscoll, Chairman of the Committee on Research, was introduced and took charge of the Research Session. In his report for the Committee he gave a resumé of the work of the Laboratory for the preceding six months, mentioning the generosity of Dean Anderson in serving as Director of the Laboratory without compensation, and telling of the efforts of the Committee to secure a Director.

Speaking of building up a reserve fund to finance the Laboratory and its work he said:

We have been operating on a mere pittance. Our income is increasing, and we are building up a reserve that is going to be of value to us as we go along. After all, we have been down to hard pan, and I think it is just as well that we have carried on as we have and established a substantial financial foundation. The money that is coming in is being spent judiciously. We didn't feel that we wanted it to go out merely because it was coming in. We didn't want to rush out and spend it, do something; we might have adopted a policy of doing something, no matter what it was, but the Laboratory isn't here for today or tomorrow. It isn't important that we do things in the month of May, 1925; it is more important that what we do is done correctly; that the things that we do are the things that we ought to be doing, and that instead of rushing out helter skelter to spend the funds that are coming in, to handle the matter as business men would handle it. That's what we are trying to do, and have tried to do.

The personnel at the Laboratory, as I say, has not been increased. It has been carried along; they have been finishing up the work they have been doing. Dean

Anderson has been keeping them busy on a number of things and in a number of ways that are intensely interesting.

Work Goes On

Now, of course, we have many suggestions come to us, many ideas are presented to us as to what we ought to do and what we ought not do, and how we should do things. One fellow says we do too much on ventilation. Not one but many many suggestions and criticisms have come in to the effect that we do too much work on ventilation. That is hardly a just criticism, because we don't do too much work on anything. We may not do enough work on some of the problems that many of our members are interested in. We hope to continue to do much of the work that Dr. McConnell has been doing, for instance, that we have been doing in conjunction with the U. S. Public Health Service. It is a very very valuable work and of the utmost importance, although perhaps, not of the greatest interest to all of our members. It is a work we can't afford to abandon, but we do hope to enlarge our program in directions that will be of particular interest to the members.

The work on critical velocities which we have been doing at the Laboratory will be continued. We have had a specific request from the *Heating and Piping Contractors' Association* who have been substantial contributors to our Laboratory, to carry that work on as it is of considerable interest to them in the work that they are doing. The question of steam and hot water radiation; this agitation of the Department of Commerce Secretary Hoover in his simplification program, has come up before us. The question as to whether or not we can recommend to the radiator manufacturers the abandonment of steam radiation, the steam type of radiator, for instance, and devote themselves to the manufacture of only the hot water type of radiator. It is a matter in which the public, the U. S. Government, is interested. It is a matter that our Laboratory can interest itself in by making a study of the question as to whether the hot water type of radiation, for instance, can be used with the same effect and the same efficiency as the steam type of radiator on steam work. If that can be done, if that can be proven, and this Laboratory I think is the place in which it ought to be proven, it will probably result in the radiator companies abandoning the manufacture of the steam type of radiator, a process of simplification that will mean a great deal to the industry of heating and ventilation. It will conserve resources, it will limit the cost of production, and that is one of the practical problems, I think, we can solve.

The question of a research residence—you saw the picture here of the splendid research residence they have at Urbana, Ill. That question is cropping up all the time—why shouldn't we have a research residence of that kind? Why shouldn't we build a residence and make a very careful study of the problem of heating the home, a study of the methods, location of radiators in the room, and all of the problems that enter into the matter? As a matter of fact, the great problem of the heating industry is heating the home. It is absolutely the most important problem before us and no definite scientific investigation of that matter has been conducted by any organization that I know of. Our Laboratory ought to be working on it.

Perhaps I am a little optimistic—but I think I can announce now that by this time next year we will have such a research residence, by this time next year I hope and I think and feel in my heart that we are going to have that residence, and we are not going to build it from funds of the Laboratory but we are going to get funds from sources apart from the normal sources of income of the Laboratory, and carry on investigations I think of the character so many of our members want. Many of our members feel that some of the investigations are entirely too scientific, that they deal with theoretical problems and theoretical conditions rather than with practical problems, and I think perhaps that may be one way to satisfy the demands of that particular group.

The Fourth Session was given over to the discussion of the Code. After discussion at some length the following resolution was offered by Thornton Lewis and unanimously adopted:

That this Society accept the printed document submitted by the Code Committee and issue it in book form as the "AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS' Compendium of Modern Practice."

More than 200 members and guests attended the Meeting. Many of those living within a short distance drove to Atlantic City in their cars.

The social program opened with an informal reception and dance held in the Rose Room at the Hotel Traymore, on Monday evening at 9 o'clock. On Tuesday afternoon a luncheon party was formed by the ladies who were taken in sight-seeing busses through Atlantic City to the Country Club of Atlantic City at Northfield. Following luncheon, tables were set for bridge. The golfers arrived at the Country Club after lunch and organized foursomes for 18-hole match play.

Visiting the Steeplechase, surf bathing and walking or riding in roller chairs along the boardwalk also furnished amusement for the engineers.

One of the outstanding events of the Meeting was a testimonial dinner given by members of the Committee on Research to Dean F. Paul Anderson on Monday evening at the Hotel Traymore. Following the dinner, Thornton Lewis presented Dean Anderson with a silver mounted malacca cane in appreciation of his services for the Laboratory.

The summer meeting was voted a great success by all who attended.

PROGRAM SEMI-ANNUAL MEETING 1925

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

Hotel Traymore, Atlantic City, N. J.

Monday, June 15, 1925

Morning—BUSINESS SESSION

11.30–1.00 P.M. Address of Welcome
Response by President Dibble
Committee Reports
Address by President S. E. Dibble
Paper: Motor Driven Return Line Vacuum Heating Pumps, by H. M. Wylie and A. D. Harvey
Paper: Eliminating Waste in Schoolhouse Planning, by F. I. Cooper.

Afternoon—RESEARCH SESSION

2.30–4.30 P.M. Discussion of plans for the Research Laboratory by the Committee on Research

Evening—

9.00 P.M. Informal Reception and Dance

Tuesday, June 16, 1925

Morning—HEATING AND VENTILATING SESSION

10.00–1.00 P.M. Paper: A Proposed Method for Comparison of Effectiveness of Indirect Heating Surfaces, by A. E. Stacey, Jr. and C. M. Ashley
Paper: Effect of Length of Leader Pipe on Heating Capacity in Gravity Warm-Air Heating, by V. S. Day
Paper: Allowance for Variations in Temperature at Register Faces for Design of Warm-Air Furnace Systems, by A. P. Kratz
Paper: Value of Fans in Furnace Heating, by C. G. Buder

Afternoon—

2.00 P.M. Golf Meet at Country Club of Atlantic City

Evening—

7.00 P.M. Entertainment at Steeplechase Park

Wednesday, June 17, 1925

Morning—CODE SESSION

10.00–1.00 P.M. Report of Committee on Minimum Requirements for the Heating and Ventilation of Buildings
Discussion led by L. A. Harding, General Chairman

SPECIAL ENTERTAINMENT PROGRAM

Monday, June 15, 1925

9.00 P.M.	Informal reception and dance, Hotel Traymore

Tuesday, June 16, 1925

12.00 M.	Automobile ride for ladies
1.00 P.M.	Ladies Luncheon at Country Club of Atlantic City followed by bridge party
2.00 P.M.	Golf Tournament for men
7.00 P.M.	Big Fun Event at Steeplechase

Wednesday, June 17, 1925

Afternoon—Rolling chair ride for ladies—Swimming

No. 728

ELIMINATING WASTE IN SCHOOLHOUSE PLANNING[1]

Some Rules Resulting from a Careful Study of 200 Structures Made by the Society's Committee on Schoolhouse Standards

By Frank Irving Cooper,[2] (Member)

BOSTON, MASS.

WHILE not presenting standards as that term is usually taken, this report does present the process of planning a school building for economy of floor space in such completeness that many of the mistakes of the past would have been avoided if there had been a clear conception by school officials, architects, and engineers of the steps that should be taken in the planning and construction of a school building.

This report has been prepared in collaboration with the Committee on Schoolhouse Planning and Construction of the *National Education Association*, which Committee originally requested the appointment of a Committee on Schoolhouse Standards by the American Society of Heating and Ventilating Engineers to cooperate in formulating standards that might receive general acceptance.

The general principles of this report apply not only to the present time but to the future as long as the planning of school buildings shall occupy the attention of architects and engineers.

Detecting Waste in the Plan

I. *Causes and nature of waste.* The chief causes of waste in the plan of a school building and the nature of these wastes are as follows:

A. Uneconomical layouts for rooms. The equipment, and aisles may be excessive in size, thereby using too many square feet of area for the number of pupils accommodated. Much scientific study is needed regarding these layouts.

B. A faulty schedule of rooms. The number of rooms needed for various purposes may not be accurately determined, thereby warping the educational program. The pupil capacity of various rooms may be made too great, thereby producing an excessive number of vacant places when the school is in operation.

[1] Report of the Committees on Schoolhouse Standards.
[2] Frank Irving Cooper, *Chairman.*
Presented at the Semi-Annual Meeting of the American Society of Heating and Ventilating Engineers, Atlantic City, N. J., June, 1925.

The pupil capacity of various rooms may be made too small, thereby either increasing the cost of instruction by requiring the division of classes, or decreasing the efficiency of instruction by requiring teachers to continually change from room to room to fit the classes to the rooms.

C. *Unwise choice of the general plan.* This may lead to poorly lighted areas or irregular angles that cannot be used economically, or to rooms or corridors with inadequate natural light and natural ventilation.

D. *Excessive areas for purposes other than instruction.* Areas for instruction includes class and recitation rooms, the gymnasium, auditorium, library and their accessories, study halls. (It also includes lunch rooms if they are used for study halls), shops, domestic arts, laboratories, and lecture rooms.

Without the thorough study and restudy of plans there is sure to be an excessive amount of floor area used for walls, partitions, flues, stairways, corridors, toilets, wardrobes, heating and ventilating apparatus, administrative offices, storage rooms, and other areas not directly connected with instruction rooms. Excessive amounts of space used for the purposes mentioned cuts down the area that can be devoted to instruction. Necessary as these provisions are, they can so be planned as not to occupy areas beyond the per cent areas allotted to them.

The main purpose is to determine the areas that may reasonably be allowed for purposes other than instruction and to indicate a method of detecting waste due to excessive areas for them.

II. *Amount of waste.* The amount of waste due to insufficient study has been estimated approximately by the N. E. A. Committee. It has been found from the measurement of over 200 school buildings that the per cent of area for instruction varies from 65 per cent or more in some buildings to 40 per cent or less in other buildings. (All of the buildings used for this investigation and tabulation were supposed to be good buildings. If admittedly poor buildings had been included the per cents for instruction would have fallen much lower.) The variation from 40 to 65 per cent means that with a given cubic content, and at a given cost, some buildings yield only two-thirds of the return in educational efficiency which they should yield. Turning it around, we can say that by careful study and restudy some architects have greatly improved the efficiency of their buildings by raising the productive areas, thereby increasing the efficiency of the plan.

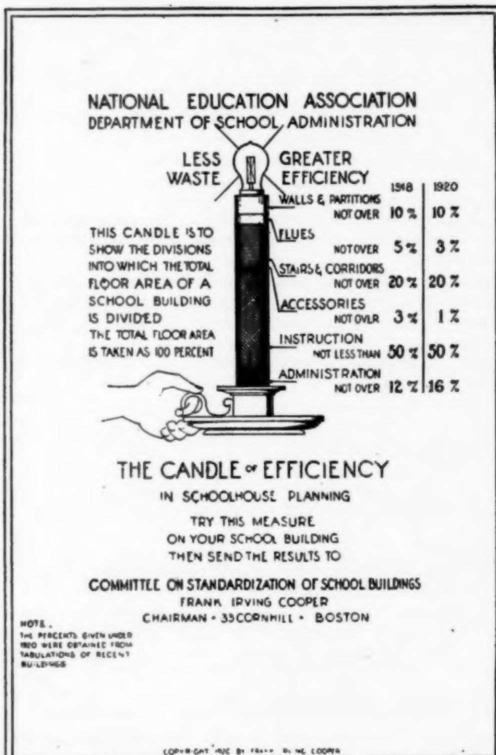
In some cases the low percentage of productive areas has been due in part to the desire of building committees to have unnecessary or unwarranted features introduced into the plan, without realizing the cost involved. In these cases it is the duty of the architect and engineer to state clearly and explicitly to their Committees the cost of these unwarranted features. In other cases the wastes are due to State regulations that are without scientific justification. In general, however, the responsibility for waste due to a low per cent of productive area belongs primarily to those directly responsible for the plan of the building and this fact should be borne in mind when individuals recommend the employment of an architect or engineer without adequate experience in planning schools.

Observation has shown many school buildings in which the layouts of instruction rooms were extremely wasteful. Not infrequently a better planned layout would have made it possible to reduce the size of the room by 20 to 30 per cent of its area without in any way cramping the work and without reducing the floor area per pupil below desirable standards. Again, buildings are built even now on the basis of a schedule of rooms that lacks balance and involves rooms with excessive pupil capacities that serve no necessary or useful purpose. The blame for bad layouts and unscientific schedules of rooms should properly be shared by the architect and the educational authorities.

The total amount of waste in school buildings due to the four combined causes mentioned is enormous. The remedy can be found only in the employment of

experts in school architecture and educational advisors who have studied their problems thoroughly.

An illustration of this lack of economy due to layouts of instruction rooms is shown in the case of two printing shops in school buildings designed by the same architect for the same city. These shops are in two different schools of the same general type and each shop is intended to accommodate twelve pupils. One has a floor area of 713 sq. ft. and the other 1050 sq. ft. If the smaller shop provides



adequate area for the successful activities of a printing class of twelve pupils in the first school then the larger shop in the other school has a waste of 337 sq. ft. The additional area in the larger shop not only represents extra cost for construction and heating, but may result in reduced efficiency because of the distances between facilities that should be close at hand. On the other hand, if all the room in the larger shop is needed, the smaller shop must be cramped. To secure the best results not only areas but also arrangements of equipment need continued study.

III. *Significance of waste.* Waste in the plan means waste in public money available for school buildings. It means continued waste in the repairs, ventilation,

heating, janitorial service, and upkeep of buildings that are needlessly large for the service they render. In many communities, waste in the plan means that the funds that these communities can and will appropriate will be exhausted before the needs of some of the children for suitable, safe, and sanitary buildings can be provided.

IV. *Method of detecting waste due to excessive areas for purposes other than instruction.* In developing its method for detecting waste due to excessive areas for purposes other than instruction the committee took the six following steps:

A. Classification of floor areas into six main divisions, five devoted to non-instruction purposes and the sixth to instruction.

B. Establishment of rules for measuring and classifying areas.

C. Selection of plans of over 200 school buildings to be measured.

D. Measurement of plans and reduction of areas to per cents.

E. Study of plans and per cents.

F. Establishment of *maximum* standards for the five non-instruction divisions and of *minimum* standards for the instruction division.

These steps are outlined in the six following sections.

A. *Classification of floor areas into six main divisions.* After extended study and conference with architects and educators the committee decided to group all the floor areas found in school buildings into the six following main divisions. They include every square foot of floor area in every building. The first five divisions are called non-instruction divisions.

1. *Stairs and corridors* includes areas that are self explanatory.

2. *Administration* includes the rooms for officials, instructors medical department, service and storage rooms, general wardrobes, sanitaries, and the area used for the heating and ventilating apparatus.

3. *Walls and partitions* includes the floor area occupied by the exterior walls and the interior partitions.

4. *Flues* includes the area occupied by the vertical flues.

5. *Accessories* includes closets, store rooms, and other areas not directly connected with the other divisions.

6. *Instruction* includes the floor areas used by all departments for purposes of instruction, including auditorium, gymnasium, and their necessary accessories. It also includes the lunch room when so planned that it can be used effectively as a study room.

B. *Establishment of rules for measuring and classifying areas.* The second step taken by the committee consisted of the establishment of the following twenty rules for the measurement and classification of areas.

1. Every square foot of floor area of the building is to be included in the tabulation. The sum of the areas of the used basement and each floor is called 100 per cent. Tabulations are to be checked until the sum of the various areas check within one-half of one per cent of the total area.

2. Line of measurement for area of all floors is to be taken at the outside of exterior walls. Deduct all recesses that are the full story height.

3. The area of basement floor is to be measured from same line as outside wall of first floor.

4. Compute each floor and mezzanine separately.

5. The area of light wells courts, open air shafts is not to be included in floor area.

6. Areas of arcades, open porches, uncovered corridors, pergolas, and open air theatres or auditoriums are to be figured separately.

7. In rooms and auditoriums which extend through more than one story the area of such space shall be deducted from the floor or floors through which it extends.

8. In the case of an assembly hall or gymnasium which has a balcony, the area of such balcony shall be taken separately.

9. In figuring walls or partition areas, no door or window openings shall be deducted, but the wall shall be figured solid, as though no openings occurred.

10. Exterior walls and interior partitions are to be figured the finished thickness, including any lath and plastering.

11. Large piers occurring in rooms are to be deducted from floor areas and added to wall areas.

12. Flues are to be figured to include all surrounding walls except walls and partitions figured under 10.

13. Chimneys are to be figured in as flue areas.

14. Where closets, bookcases, or dead spaces occur in a bank of flues, same are to be figured in as flue area.

15. Stairs extending a full story in height and stairs leading to a mezzanine story are to be taken as stair area. Steps not a full story in height are to be taken as part of the floor area of the room or corridor in which they occur.

16. Area of each individual space is to be taken separately in accordance with schedule.

17. Wardrobes are figured inside the walls.

18. Rooms having wall cabinets are to be figured from partition walls. Cabinets are to be included with the room.

19. Waiting spaces, closets, supply rooms, toilets, etc. in connection with offices shall be included with the office with which they occur.

20. Toilets, showers, storage rooms, supply rooms, etc. when connected with a main division, shall be taken separately but be included with the total area of that division.

C. Selection cf plans of over 200 school buildings to be measured. The first collection to be measured consisted of the plans of about 110 school buildings. They were all selected from the work of architects of such standing in their profession as to command public confidence.

They included buildings of many types such as: buildings of one, two and three stories, buildings with and without basements, and buildings with and without auditoriums and gymnasiums.

Some of the schools were situated in rural districts, some in small towns and cities, and some in the largest cities, of the United States. They included kindergartens, elementary schools, junior and senior high schools, and vocational industrial and trade schools.

This set of 110 plans was reduced to 80 plans by the eliminations of those plans that seemed to have special features that would make their inclusion misleading, or that were erratic. Kindergartens and vocational, industrial and trade schools were also excluded.

The 80 schools finally retained in the group included only elementary schools and junior and senior high schools. They were distributed geographically by 24 states as follows:

California	1	Nevada	1
Connecticut	2	New Hampshire	1
Delaware	1	New Jersey	4
Florida	6	New York	6
Illinois	8	North Dakota	1
Indiana	1	Ohio	8
Maryland	1	Oregon	2
Massachusetts	11	Pennsylvania	7
Michigan	2	Texas	2
Minnesota	8	Utah	1
Missouri	3	Vermont	1
		Washington	1
		Wisconsin	1
			80

Of these 80 schools

35 were elementary schools
6 were junior high schools, and
39 were high schools including consolidated schools.

After the study of the first set of 80 plans it was thought that the results might be modified if additional schools were tabulated. Accordingly, the committee secured about 100 additional plans. But the measurement of these schools did not show materially different conditions, except that some of the more recent schools revealed more careful planning, induced no doubt in part at least by the work of the committee in pointing out sources of waste.

D. Measurement of plans and reduction of areas to per cents. To obtain the percentage of floor area in each of the main divisions, the entire floor area of the building was taken as 100 per cent. The area of every room, closet, corridor, stairway, flue and wall was measured. These areas were grouped under the six main divisions. If the sum of all the areas did not tally within one-half of one per cent of the total floor area, the error was searched for and the tabulation was not used until the sum of the items was brought within one-half of one per cent of the total area. The per cent of the floor area of each main division to the total floor area was then found.

E. Study of plans and per cents. For the purpose of comparing the various per cents obtained from measuring the plans, a little strip of cardboard, 1 in. wide and 22 in. long was made for each building. Twenty inches on each strip represented 100 per cent and on this 20 in. length there was laid out to scale the per cents found for each of the six main divisions. These strips, which looked like candles, were brought together as desired for purposes of comparisons and they were used in laying out the various charts showing the results of the study.

Some statisticians claimed that there could be little similarity between the areas of elementary and high school buildings, and that consequently it would be necessary to establish separate standards for them. This claim was not substantiated.

F. Establishment of maximum standards for the five non-instruction divisions and of minimum standards for the instruction division. On the basis of the measurement and study of the 80 buildings retained from the first set of 110 plans, the committee in 1918 established its first set of maximum and minimum standards.

Further study led the committee to revise these standards. The revised standards are given in the form of a candle and has been called by the committee the *Candle of Ratios*. The name *Candle of Ratios* was suggested by the little strips referred to in the preceding section.

The sum of the percentages allowed by the committee for the first five divisions equals 50 per cent. Hence a building in which this sum does not exceed 50 per cent will meet the sixth standard of the committee, that is, it would have 50 per cent or more for instruction.

Suppose now that a building meets every one of the first five standards, and that it shows considerable more economy than is demanded in at least some of the standards. Then the available space for instruction will rise considerably above 50 per cent.

V. *Results from the study of 80 buildings.* As already stated these 80 buildings upon which this study is based were selected from 110 plans by rejecting plans that seemed misleading or erratic.

The plans were gathered from 24 states as indicated in the preceding section. They were all from architects of recognized standing.

These 80 schools included:

35 Elementary Schools,
6 Junior High Schools, and
39 High Schools (including Consolidated Schools).

Junior high schools and high schools, as groups, succeeded much better than did elementary schools in meeting the standard of 50 per cent for instruction. The superiority of the junior high schools is probably due to the fact that they are of a more recent type. The superiority of both high schools and junior high schools is probably due to the fact that they average larger buildings, that a larger percentage of them had auditoriums and gymnasiums, and that they were planned more carefully. It is evident, moreover, from the fact that at least 40 per cent of each of the three groups met the standard and at least 33 per cent did not and that the median

for each group was near the standard, that there is no need for separate standards for instruction for the three groups.

In the five non-instruction divisions it was found that elementary schools succeeded best in reducing flues and accessories, junior high schools in reducing stairs and corridors and walls and partitions, and high schools in reducing administration.

The per cents value showing the greatest uniformity for all schools is the one for walls and partitions, or construction as it may be called. This is accounted for because construction is governed by building codes.

Of the per cents devoted to instruction the minimum standard is 50 per cent. Forty-two schools fell below this minimum standard, the lowest devoting only 37.53 per cent to instruction. Thirty-eight schools do better than the minimum standard, the school best in this respect devoting 67.95 per cent to instruction.

Heating and ventilating, ranging from $1^1/_2$ per cent to 15 per cent shows an excessive variation, indicating clearly divergent views as to the proper methods of ventilation. These per cent values should be of especial note to the members of the Society.

Wardrobes range from $^1/_2$ to 7 per cent. This variation also indicates either that some schools do not provide proper care for clothing, or that other places do not use economical methods.

Storage ranges from 0 to 5 per cent. In some cases the architect marked storage on waste spaces that he seemed unable to utilize otherwise.

Toilets range from 1 to $6^1/_2$ per cent.

Offices and teachers' rest and emergency rooms range from $^1/_2$ to 6 per cent.

These findings indicate clearly the need for the study of details in planning a school building and for scientific standards based on actual requirements.

In the study of the subdivisions of instruction in 47 departmental schools, all but 11 of these schools were from among the 80 schools discussed in this section. The variations in the different subdivisions were as follows:

For academic rooms from 9.5 to 36.5 per cent.
For laboratories from 0 to 14.4 per cent.
For industrial shops from 0 to 30.8 per cent.
For agriculture from 0 to 4.5 per cent.
For commercial rooms from 0 to 15.5 per cent.
For libraries from 0 to 2.8 per cent.
For household arts from 0 to 4.0 per cent.
For social activities including gymnasium and auditorium, from 0 to 31.5 per cent.

The following facts appear from a study of the 80 school buildings before mentioned:

17 either met every standard or every standard but one.

All but 8 of the schools that fail to meet the minimum standard for instruction exceeded the maximum standards for three or more non-instruction divisions.

10 met every standard except *administration.*
2 met every standard except *stairs and corridors.*
1 met every standard except *accessories.*
1 met every standard except *walls and partitions.*
9 met every standard except *administration* and *walls and partitions.*
4 met every standard except *stairs and corridors* and *walls and partitions.*

VI. *Results to be gained from tabulating the floor plans.*

1. The tabulation will aid persons interested to translate the plan into a problem in economics.

2. It is one of the best methods of detecting a kind of waste that used to be found in most school plans and that is still found in plans that are not carefully studied.

3. It will show the problem divided into its simplest units and will enable the persons interested to at once point out what gains can be made through a restudy of the plan.

4. It will arouse the interest of every person who studies the plan as a whole. He will appreciate the waste that may exist in a seemingly excellent plan and will see the actual benefits to be accomplished by a restudy.

The fact that 50 per cent or more of the total area of a plan is used for instruction does not prove that there is no waste within the instruction areas. The rooms themselves may be too large for the purposes for which they are intended. The aisles may be unnecessarily wide or badly designed and the equipment may be too large, or the room may be designed for a larger number of pupils than will or should occupy the room. To eliminate these sources of waste it is necessary that thorough studies should be made as to desirable and economical layouts and to the proper sizes of classes in different grades of the elementary school and different subjects in the high school.

If, on the other hand, the area for instruction does not equal or exceed 50 per cent, or if the areas for any of the other divisions is much in excess of the percentages given by the committee, then it is almost certain that there is waste and the plan should be restudied.

The use of the per cents of floor areas for purposes of judging and restudying plans as applied to school architecture is new. The method has been developed by the Committee on Schoolhouse Planning and Construction appointed by the *National Education Association.*

This method consists of measuring the square feet of the main divisions of the plan from architect's drawings and changing these areas into per cents of the entire floor area. If there is lack of economy in the plan these per cents will show where departure from the standards occurs.

Many educators and architects still plan school buildings as they did before the war showing that the day of the "rule of thumb" is not yet ended. Many of them give little study to the proportional distribution of areas for the various departments. They apparently give much time and attention to relatively trivial elements while more serious concerns pass unnoticed.

Many an architect engaged in the preparation of plans for a school building is content with his plan if his committee seems satisfied. This type of architect seldom studies what goes on in the school building after it leaves his hands. He knows that a certain number of rooms are required; that there must be ways to approach them; that for so many pupils certain plumbing fixtures must be provided; that there must be stairways for exit; clocks to tell time; telephones to give communication; a heating plant to warm the building; and ventilating ducts to satisfy building codes. Oftentimes this architect is a truly educated man, the design of his building merits praise and he employs experts to arrange for the engineering aspects of his work. He particularizes when he knows; he generalizes and is vague when he does not. He knows the art of design and his committee is pleased with his work; the educator knows he wants so many rooms, he is given them and is content; the inspectors insist that certain technical requirements be met and the experts of the architect's organization meet these requirements and the plans are passed; but neither the architect nor the educator knows whether there is waste in the plan.

When plans are studied and measurements and tabulations are made of the

floor area in accordance with the rules presented in this study, the per cents of the main divisions may be easily read and comparisons made. By studying the details in the tabulations, the superintendent or business official may learn in what division waste and lack of economy exist. If waste does exist, he will have positive data from which to suggest a restudy of the plan.

Tabulation of the floor areas according to the rules, comparison of the results, a restudy of the plan if the per cents show large departures from the standards and especially if the per cent for instruction is low, will contribute to securing a more successful and economical building.

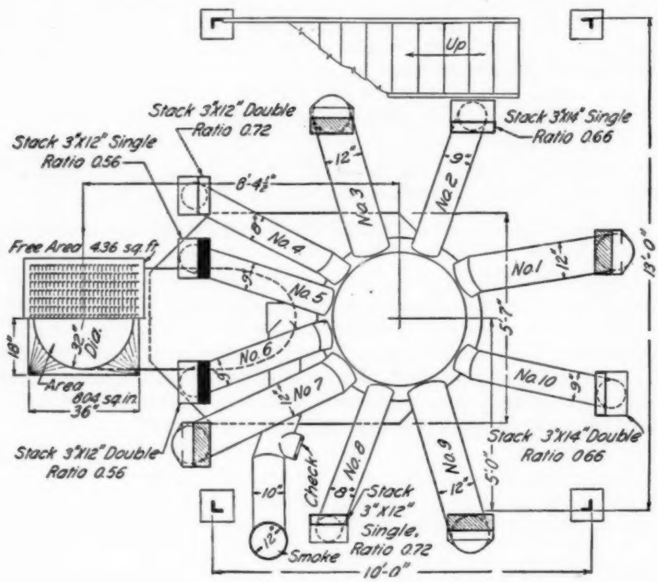
DISCUSSION

H. W. WHITTEN: I would like to ask if a record of the kinds of heating plants in these buildings were kept, and were they all of a modern ventilated type?

H. R. LINN: Mr. President, I would like to ask Mr. Cooper, if he said that as low as 28% of the floor space in some buildings were used for instruction. What was the highest amount?

FRANK IRVING COOPER: In connection with the question as to the kinds of heating and ventilating systems I would say that no record was kept. All of these methods in all of these buildings were of some form of steam heating apparatus, and because of the type of buildings which were tabulated, all probably had forced ventilation of some type. In connection with the other question, the lowest per cent was 37 and the highest per cent was 68.

Allowance for Variations in Temperature at Register Faces for Design of Warm-Air Furnace Systems



	Leaders			Stacks		Registers	
	No.	Size	Area	Dimensions	Type	Dimensions	Free Area
1st. Floor	1	12 in.	113 sq. in.	———	Asbestos Covered	$11\frac{3}{16}$ in x $12\frac{3}{8}$ in	0.684 sq. ft.
	3	12 in.	113 sq. in.	———		" "	"
	7	12 in.	113 sq. in.	———		" "	"
	9	12 in.	113 sq. in.	———		" "	"
2nd. Floor	2	9 in.	64 sq. in.	3 in x 14 in.	S'gle.	$8\frac{1}{8}$ in x $11\frac{1}{8}$ in.	0.458 sq. ft.
	4	8 in.	50 sq. in.	3 in. x 12 in	D'ble.	" "	"
	8	8 in.	50 sq. in.	3 in. x 12 in.	S'gle.	" "	"
	10	9 in.	64 sq. in.	3 in. x 14 in.	D'ble	" "	"
3rd Floor	5	9 in.	64 sq. in.	3 in. x 12 in.	S'gle	" "	"
	6	9 in.	64 sq. in.	3 in. x 12 in.	D'ble	" "	"
Leader Area: 1st. Fl. 452 sq. in., 2nd 228 sq. in., 3rd. 128 sq. in. Total 808 sq. in.							
Percent Leader Area: 1st. Fl. 55.9, 2nd. 28.2; 3rd. 15.8.							

Fig. 1. Plan and Dimension Table for Piped Furnace Test Plant

No. 729

ALLOWANCE FOR VARIATIONS IN TEMPERATURE AT REGISTER FACES FOR DESIGN OF WARM-AIR FURNACE SYSTEMS

A. P. KRATZ,[1] URBANA, ILL.

MEMBER

Introduction

THE data used in this report were taken from tests forming a part of the program of the investigation of warm-air furnaces conducted by the Engineering Experiment Station at the University of Illinois under a cooperative agreement with the *National Warm-Air Heating and Ventilating Association*, and directed by A. C. Willard, professor of heating and ventilation and head of the department of mechanical engineering.

Description of Plants

A plan of the plant used in the Mechanical Engineering Laboratory is shown in Fig. 1. The size and character of the leaders, stacks and registers are shown on the plan and in the table. The length of the leader pipes is given in Fig. 3. It should be noted that in this plant the leaders are comparatively short and are practically all of the same length, approximately 4.0 ft.

A plan of the plant used in the Warm-Air Research Residence, which consisted of a typical residence built for the purpose of conducting investigations of warm-air furnace systems, is shown in Fig. 2. The length of leaders used in this plant may also be obtained from Fig. 3. In this plant the length of leader pipe was determined by the location of the room it supplied, and it varied from a minimum of 1.7 ft. to a maximum of 11.5 ft.

A 27-in. cast-iron circular-radiator furnace was used in both plants. This furnace had a 23-in. grate and was cased with a 52-in. casing having a black iron inner lining spaced one inch from the outer casing and extending from the top of the cold-air shoe to the bottom of the bonnet. With the exception of the length of leaders the two plants were as near similar as possible.

Discussion of Results

The curves in Fig. 3 show the distribution of the temperature of the air at the register faces for the two plants tested, for combustion rates of approximately 6 lb.

[1] Research Professor, Department of Mechanical Engineering, University of Illinois.

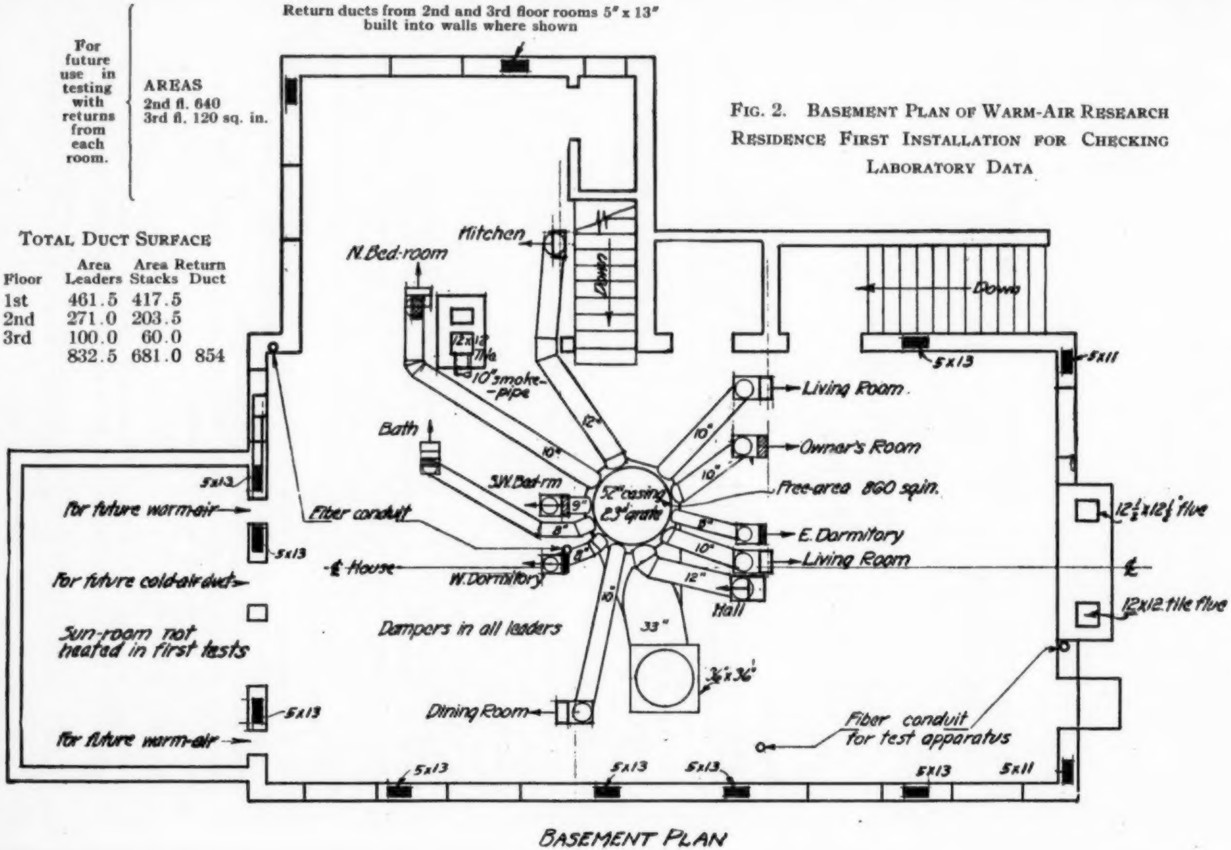
Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Atlantic City, N. J., June, 1925.

FIG. 2. BASEMENT PLAN OF WARM-AIR RESEARCH RESIDENCE FIRST INSTALLATION FOR CHECKING LABORATORY DATA

For future use in testing with returns from each room. AREAS 2nd fl. 640 3rd fl. 120 sq. in.

TOTAL DUCT SURFACE

Floor	Area Leaders	Area Stacks	Return Duct
1st	461.5	417.5	
2nd	271.0	203.5	
3rd	100.0	60.0	
	832.5	681.0	854



of coal per square foot of grate per hour. The radiating lines represent the position of the leaders at the bonnet of the furnace and the concentric circles represent the scale of temperatures. The connecting lines between points on the various leaders have no significance further than serving to emphasize the shape of the diagram. That is, there is no reason to believe that if a leader were installed between any two leaders shown, the temperature would be given by the intersection of the position line and the line joining the temperature for the two adjacent leaders. This would be true only in case the diagram indicated some definite law for the variation in the temperature around the plant. The existence of such a law is not evident.

Fig. 3 shows that while the average temperature of the air at the register faces was practically the same for both plants, 156 deg. fahr. for the laboratory plant and 159 deg. fahr. for the one in the Research Residence, the deviations from the mean were very material. Furthermore, while there does not seem to be a definite law for the temperature variation around the plant there was a marked tendency for the higher temperatures to be found at the rear of the plant. This was true for

DIMENSIONS OF LEADERS, STACKS AND REGISTERS OF RESEARCH RESIDENCE

Floor	Room	Leaders Dia.	Leaders Area	Stacks (or throat) Size	Stacks Type	Stacks Area	Registers Size	Registers Free Area	% Free
First	Living	10″	78.5	$5^3/_8$ x $12^7/_8$*	D	71.5	10 x 12	83.5	70
	Living	10″	78.5	$5^3/_8$ x $12^7/_8$*	D	71.5	10 x 12	83.5	70
	Hall	12″	113	$7^1/_2$ x 14	S	10.5	12 x 14	120.5	72
	Dining	10″	78.5	$5^3/_8$ x $12^7/_8$*	D	71.5	10 x 12	83.5	70
	Kitchen	12″	113	7 x 14	S	98	12 x 14	120.5	72
Second	Owner's	10″	78.5	5 x 12	S	60	10 x 12	83.5	70
	S. W. Bed	9″	64	$3^1/_2$ x 12	S	42	9 x 12	74.0	69
	Bath	8″	50	3 x 10	D	30	8 x 10	53.0	76
	N. Bed	10″	76.5	$5^3/_8$ x $12^7/_8$	D	71.5	10 x 12	83.5	70
Third	E. Dormitory	8″	50	3 x 10	S	30	8 x 10	53.0	67
	W. Dormitory	8″	50	3 x 10	D	30	8 x 10	53.0	67

* As catalogued, but actually $5^1/_2$″ x 13″.

both plants, although the variation was somewhat modified by the length of leaders in the plant in the Research Residence. Inspection of Fig. 3 will also show that approximately the same variations around the plant occurred independently of whether first, second or third floor leaders were used in the corresponding positions. Apparently, therefore, the temperature of the air in the bonnet is not uniform, and the initial temperature of the air going to the several leaders is a function of the type of furnace, or of its orientation relative to the plant, rather than of the character of the plant itself. The actual temperature at the register faces is determined by a combination of this factor and the temperature drop in the leaders and stacks.

From Fig. 3 it is evident that if a plant is designed on the basis of some uniform temperature at the register faces, and the capacities of the leader pipe calculated such that at this register temperature they will compensate for the heat losses from the various rooms, the plant will not be balanced when it is put into operation. That is, if the doors are closed between rooms, some rooms will receive less heat than required and others may receive more, since the temperatures at the register faces will be different from that assumed, and the heat carrying capacities will vary to correspond to the register temperatures. It will then be necessary to equalize the temperature in the rooms by leaving all doors open, or by adjusting dampers in the leader

pipes. The latter procedure is not to be recommended as it reduces both the capacity and efficiency of the plant as a whole.

This may be illustrated by the case of leader No. II in the Warm-Air Research Residence shown in Fig. 3. This plant was designed on the basis of 175 deg. fahr. at the register faces with zero deg. fahr. outdoors. Fig. 3 shows that two registers, namely No. VIII and No. X, operated at this temperature when the outdoor

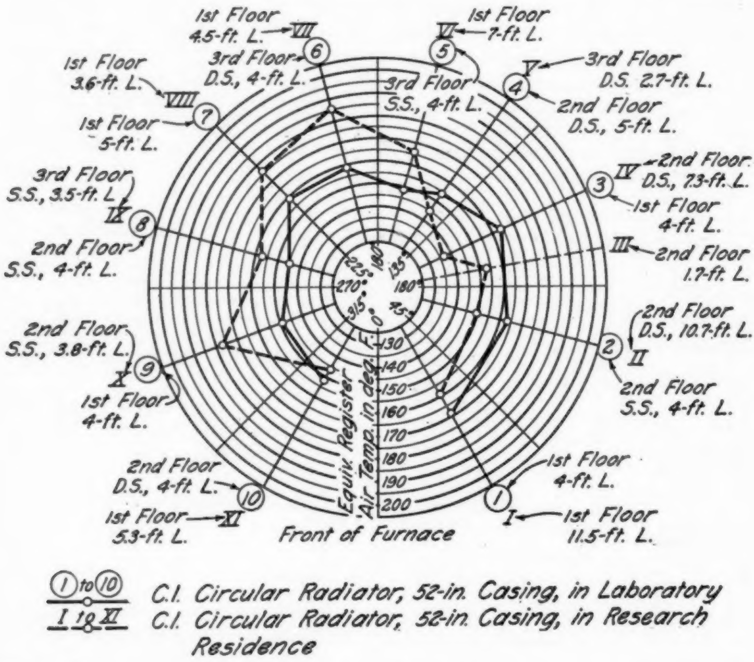


FIG. 3. DISTRIBUTION OF TEMPERATURE AT REGISTERS—EQUIVALENT TEMPERATURE ABOVE 65 DEG. FAHR. INLET IN LABORATORY AND RESEARCH RESIDENCE PLANTS

temperature was 19 deg. fahr. Register No. II operated at a temperature of 150 deg. fahr., or 25 deg. fahr. lower than the 175 deg fahr. shown by leader No. VIII. The heat loss from room No. II was 13,850 B.t.u. per hour with zero outdoor temperature. If this were supplied at a register temperature of 175 deg. fahr., it is evident that the temperature at the register face in room No. VIII would be much higher than 175 deg. fahr. Notwithstanding these individual variations, however, the average register temperature at the register faces in the Research Residence was practically 175 deg. fahr. for zero weather.

For the purpose of this discussion, since it is intended to show what bearing variations in register temperature have on the design of a plant only, let it be assumed that the plant from which Fig. 3 was obtained had operated such that the

heat loss from room No. VIII on a zero day was just offset by the capacity of leader No. VIII operating with a temperature of 175 deg. fahr. at the register face. Let it also first be assumed that all registers including No. II are to operate at 175 deg. fahr. on this day. The curves for leader carrying capacities given in Fig. 4 are reproduced from Engineering Experiment Station Bulletin No. 141. These show that for a second floor leader with the register temperature of 175 deg. fahr. assumed

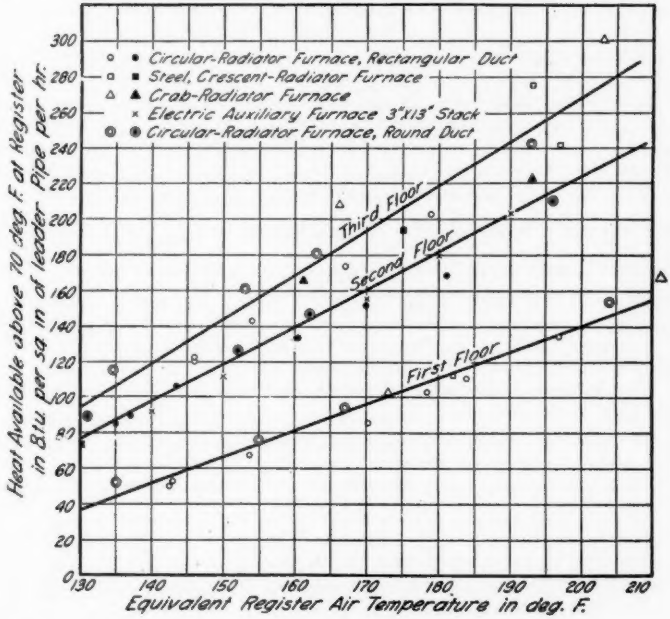


FIG. 4. VALUE OF SQUARE INCH OF HEADER PIPE AREA FOR FIRST, SECOND AND THIRD FLOORS

for a zero day, the heat carrying capacity per sq. in. of leader area is 170 B.t.u. per hour. On this basis a leader having an area of $\frac{13,850}{170} = 82$ sq. in. would be required. A 10-in. leader having an area of 79 sq. in. was actually installed. Fig. 3 shows that this would operate at a register temperature of 150 deg. fahr. instead of at the 175 deg. fahr. assumed, and at this temperature the carrying capacity from Fig. 4 is 120 B.t.u. per sq. in. of leader pipe per hour instead of 170. Accordingly on a zero day instead of supplying the 13,850 B.t.u. per hour required it would supply $79 \times 120 = 9500.0$.

If it were desired to design a perfectly balanced plant, the temperature allowable at some one, or more, registers would have to be assumed as a base. If, for this example, the operating temperature of 175 deg. fahr. at the register face with zero deg. fahr. outdoors is again assumed for leader No. VIII then leader No. II would have to be designed to give its required capacity when operating at some other

temperature, which is to be determined. The lower operating temperature of 150 deg. fahr. in this particular example was caused by (1) temperature drop in the part of the leader in excess of the 4 ft. length of leader No. VIII (2) position of the leader on the bonnet. The length of leader No. II is practically 11 ft. The average drop in temperature per ft. of leader has been found to be approximately 2 deg. fahr. Hence a drop of $(11 - 4) \times 2 = 14$ deg. fahr. greater than that for leader No. VIII would be expected for No. II. Sufficient data to enable the direct determination of the effect of the position on the bonnet for all cases is not at present available. In the case under discussion, however, it may be approximated as 11 deg. fahr. That is, the air enters leader No. II at a temperature 11 deg. fahr. lower than that for which it enters leader No. VIII. The sum of these two factors is, therefore, $11 + 14 = 25$ deg. fahr. Accordingly it would be expected that when the temperature at the register face for leader No. VIII was 175 deg. fahr., that at the register face for leader No. II would be $175 - 25 = 150$ deg. fahr. The curves in Fig. 4 show that the leader carrying capacity at 150 deg. fahr. register temperature is 120 B.t.u. per sq. in. per hour. In order to supply the heat loss of 13,850 B.t.u. per hour the area of leader pipe No. II would have to be $\frac{13{,}850}{120} = 115$ sq. in. The nearest size of standard pipe would be 12 in. In order to design a properly balanced plant this procedure must be followed for each leader.

Conclusions

The following conclusions may be drawn.

1. The distribution of the temperature at the register faces of a warm-air furnace plant is not uniform.

2. The unequal distribution of the temperature at the register faces is caused by (*a*) unequal temperature drops in the leaders, (*b*) position of the leader on the furnace bonnet, or unequal distribution of the temperature of the air in the bonnet.

3. The distribution of the temperature of the air in the bonnet is independent of whether first, second, or third floor leaders are used at the different points, but is determined by the character of the furnace used, or by the orientation of the furnace with respect to the plant.

4. In general the temperature of the air in the bonnet tends to be highest at the rear of the furnace.

5. In designing a properly balanced warm-air heating plant account should be taken of the variation in the temperature of the air in the bonnet as well as of the variation in the temperature drops in the several leaders.

No. 730

EFFECT OF LENGTH OF LEADER PIPE ON HEATING CAPACITY IN GRAVITY WARM-AIR HEATING SYSTEM

V. S. Day, Urbana, Ill.

MEMBER

As PART of a series of studies of pipes and fittings for gravity warm air heating a previous report dealt with the effects of stack size and of the ratio of stack area to leader area on the capacity of pipes for heating.[1] This report, the second of the series, deals with the effect of the length of the leader or basement pipe on the heat delivered. Two general cases are discussed:

a. Leaders connecting with second story pipes.

b. Leaders connecting with first story baseboard registers.

Apparatus

The principal features and details of the electrically heated testing furnace have been described in the previous report.[1] The furnace consists of a heavily insulated electrically heated drum, provided with only one inlet pipe and one outlet pipe and equipped with instruments for the accurate determination of the quantity of air flowing and the temperatures at various points in the system.

In the tests with which this report deals, the pipe length was varied by inserting sections. Fig. 1 shows the furnace and a long pipe connected. Figs. 2 and 3 show details of the combinations tested. It is important to note that the elevation of the register, boot, and furnace with respect to each other was not changed. Consequently the slope of the pipe varied as the length of leader was varied. It is also important to note that the stack area was 72 per cent of the area of the leader.

Test Results

The test data and results are shown graphically in Figs. 4, 5, 6 and 7.

In Fig. 4 the most striking results are shown. In this case the heating effect at the registers for the seven arrangements of leader pipe, shown in Figs. 2 and 3 were plotted against *register air temperature*. The points representing tests on the leaders with *second floor* stacks connected (A, B, C and D) lie along one line, indicating that within the range of lengths tested *the heating effect is not dependent on leader length* if a constant register air temperature is maintained.

---

[1] Journal, A.S.H.&V.E., July, 1923.

Presented at the Semi-Annual Meeting of the American Society of Heating and Ventilating Engineers, Atlantic City, N. J., June, 1925.

In the case of the first floor pipes, however, some difference in heating effect for various lengths of stack were found. The curves E, F and G, Fig. 4 show considerable spread. The percentage decrease in heating capacity due to long pipes to first floor rooms amounted to about 1 per cent per foot of length. This is based on a fixed register air temperature.

Fig. 5 shows the effect of maintaining a fixed rate of heat input to the furnace and allowing the air quantity and air temperature to vary. The full lines A, B, C and D represent the effect for various lengths of pipe to second story rooms, and the dotted lines E, F, and G for the first story rooms. Unlike the curves of Fig. 4

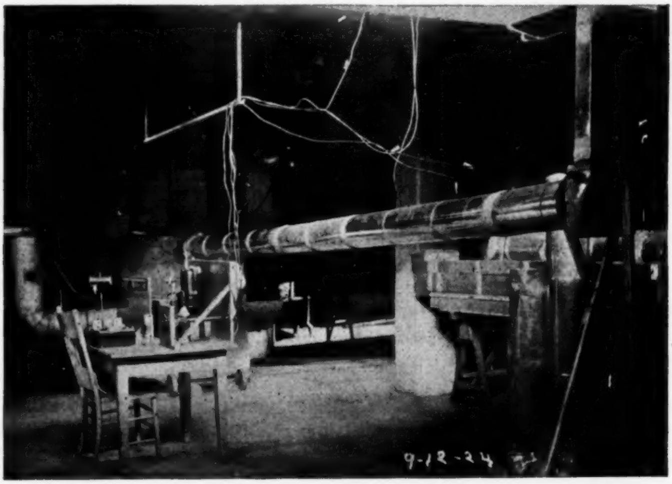


FIG. 1. VIEW OF ELECTRICALLY HEATED FURNACE AND LONG LEADER PIPE

which were based on fixed register-air temperature, these curves based on fixed heat input to the furnace are widely separated.

Manifestly, this great difference in the heating effect obtainable with various lengths of leader is caused by the loss of heat from the lengths and surface areas exposed. The curves of Fig. 7 show the drop in temperature in the various lengths of leader pipes at three register temperatures. It is important to note that the magnitude of these temperature losses was as great as 50 deg. fahr. between bonnet and boot, and conversely, were as small as 3 deg. It is also important to note that the temperature losses were less in 8-in. first floor leaders than in the same lengths of 8-in. second floor leaders. This may be accounted for by the evidence in Fig. 4 that the first floor pipes do not handle as much air as the second floor pipes and do not *run full* of air. Thus the bottom surfaces of the first floor pipes are so much cooler than the upper surfaces that the mean temperatures of the pipe surface is reduced and a smaller drop in temperature in the leader pipe results.

The identity between curves A, B, C, and D of Fig. 4, may be accounted for by the increased motive head available to overcome the greater friction of the longer pipes. That a greater motive head existed can be proved since higher temperatures

would exist in the lower part of the systems having the longer pipes, if the hypothetically constant register delivery temperature was maintained. The identity simply means that in A, B, C, and D of Fig. 4, the ratio of motives head to total resistance was constant.

This identity in A, B, C and D curves and the variation in E, F and G curves of Fig. 4 may be accounted for by the fact that the second floor systems were restricted by a 28 per cent reduction in area of the stacks as compared with the leaders, whereas the first floor systems were not restricted at boot, throat, or register face. Thus the controlling resistance in the second floor systems *was not* the resistance

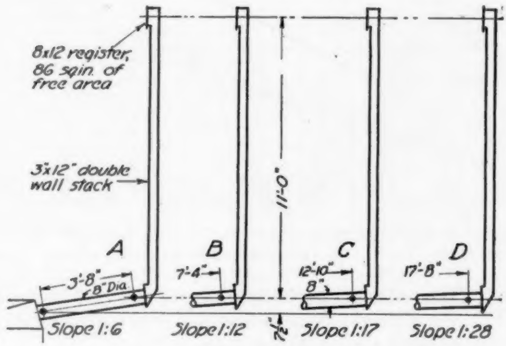


FIG. 2. DETAILS OF SECOND STORY PIPES TESTED

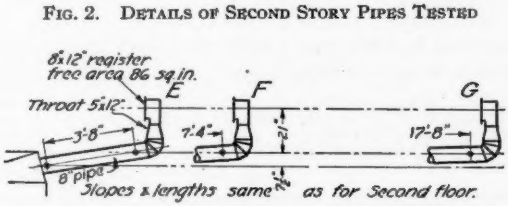


FIG. 3. DETAILS OF FIRST STORY PIPES TESTED

of the leaders but that of the stacks, and the controlling resistance in the first floor systems *was* the resistance of the leaders.

Referring again to Fig. 5 and the wide variation in heating effect on the basis of fixed heat inputs, it seemed desirable to obtain from these data a graph of the relation between length of leader pipe and heating effect at the registers, and this graph is shown in Fig. 6. Here heating effect was plotted against pipe length for various rates of input. The curves indicated are straight lines within the range of the tests, and show that heating effect varies inversely with leader pipe length. The amount of the loss in heating effect is approximately 1.5 per cent per foot of length. Thus a pipe 12 ft. long would deliver 10×1.5 or 15 per cent less heating effect at the register than a pipe of negligibly short length.

An analysis of the temperature loss in leaders is shown in Fig. 7. When the temperature drop in the leaders was plotted against length of leader, it was shown that straight line relations existed. The temperature drop was directly propor-

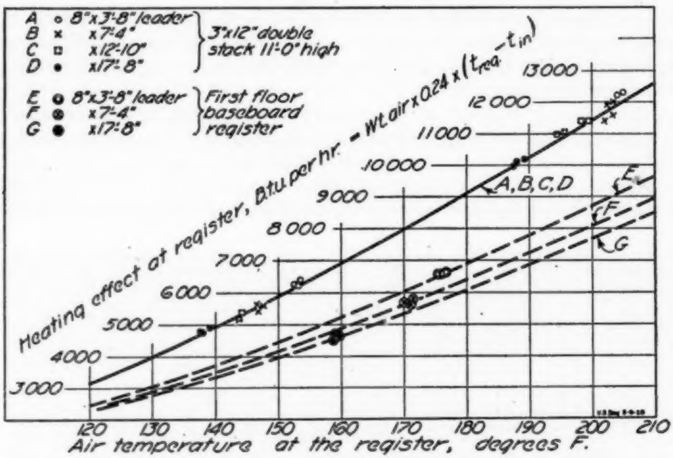


FIG. 4. HEATING EFFECT OBTAINED WITH VARIOUS LENGTHS OF FIRST AND SECOND STORY PIPES FOR VARIOUS REGISTER-AIR TEMPERATURES

tional to length. Thus, referring to the upper curve of Fig. 7 it may be seen that at a length of 5 ft. the temperature drop was 12.5 deg. and at double the length, 10 ft., the drop was doubled also, or 25 deg. fahr.

Conclusions

The data presented in this report support the following conclusions, some of which were drawn in the previous report, that:

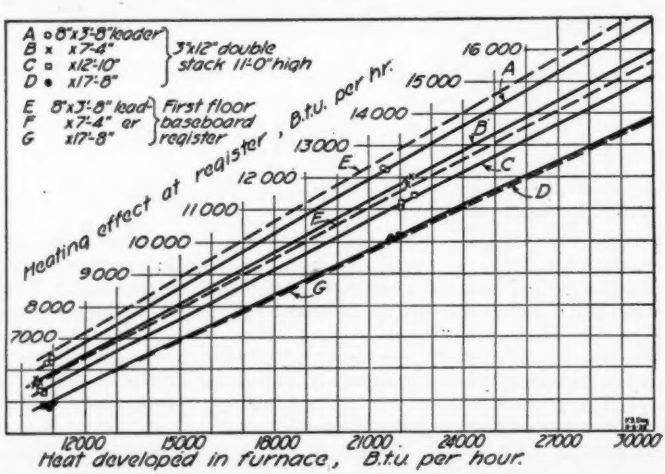


FIG. 5. HEATING EFFECT OBTAINED WITH VARIOUS LENGTHS OF FIRST AND SECOND STORY PIPES FOR VARIOUS FURNACE INPUTS

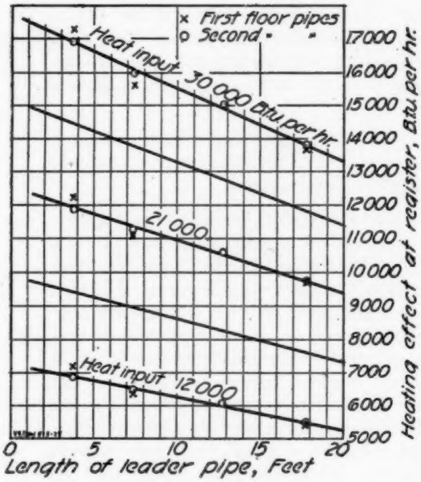


FIG. 6. RELATION BETWEEN LENGTH OF LEADERS PIPE AND HEATING EFFECT FOR SEVERAL RATE OF HEAT INPUT TO FURNACE

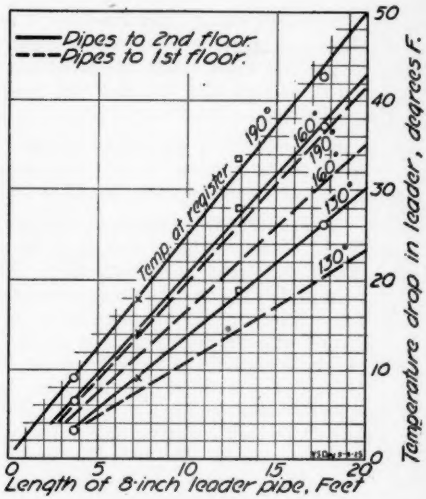


FIG. 7. TEMPERATURE LOSS IN LEADER PIPES AT VARIOUS TERMINAL OR REGISTER AIR TEMPERATURES

1. The resistance of the leader pipe is of less importance in the determination of heating effect obtainable than the resistance of the stack.

2. The pitch of the leader pipe is not an important factor in the heating effect of pipes having fixed heights between furnace and registers. This should not be interpreted to mean that a steeply pitched pipe will not heat up more quickly from the cold state than a pipe of low pitch.

3. It is more important to make allowance for length of leader pipe to first floors than to second floors.

4. The temperature losses in long lengths of leader pipe are so great as to warrant the use of special insulation of very long runs.

5. In designing systems on the basis of register-air temperature a uniform temperature should not be assumed to exist at the registers if much variation in leader pipe length exists. Instead the individual pipe sizes should be based on their respective and correct register-air temperatures.

JOINT DISCUSSION

A. S. ARMAGNAC: In direct heating work, the instructions are always given to locate the radiators directly under the windows, and even in indirect heating work where the radiators are located below the floors the registers are almost invariably on the outside walls. In the case of this furnace heating design, the first installation that has been made in the Research Laboratory, the leaders, following the usual custom, are made as short as possible, bringing the registers all at an inside location. I want to ask whether plans are being made to have installations tested with leaders running to the outside walls.

J. A. DONNELLY: Mr. President, I would like to ask Mr. Day whether the influence of open fire-places and the use of open fire-places with various fuels, such as gas and wood, which are easily kindled, on furnace heating has been investigated with the idea that the open fire-place as an adjunct to any heating system has its place.

H. R. LINN: I would like to ask Prof. Day what they found in this research residence in reference to the re-circulating of air. I believe they can re-circulate from each room and from the hall or take it all from the outside.

V. S. DAY: It is planned to run the leaders to outside walls, install the registers in the outside walls as well as in the floors near the outside walls and thereby obtain some data on that question. We have in the residence one large reception room which has an open fire-place, and we expect to determine the effect of having the damper in that fire-place open, upon air changes. We are not greatly concerned with the burning of fuel in the fire-place, and we do not expect to do any particular amount of work on that. We are more concerned with the effect of the fire-place and the chimney upon the draft of other chimneys adjoining. The research residence has five flues of various constructions or various locations. We have flues built entirely inside of the cubic of the building, and we also have flues entirely exposed. We have tile lined flues; we have common brick construction flues and we have 8 x 12 flues and 12 x 12 flues. From that group of five flues we expect to get some definite information about draft. We are interested in what effect the fire-place flue is going to have upon the two adjacent flues, particularly if there should happen to be a breakage of tile or spawling off of the tile, which will constitute a stoppage. Answering Mr. Linn's question, we have done nothing on the re-circulation of air from the individual rooms. There is provision for making each room in the house a circulating unit in itself. The house is built with cold air ducts laid in the outside walls, two or three of them, from each room, 14 of them, altogether, so it will be possible to connect them with the furnace and re-circulate each room absolutely separate as a heating unit in itself. That was not done this past Winter—we have had the house in operation since December 3rd, which is the time it was dedicated and turned over to us for use. During the first Winter's work we ran it entirely with the plant which I showed you on the screen, one large re-circulation duct located centrally in the hall, the idea being to obtain the information in stages. We expect to devote one whole Winter to each type of installation and get a variety of weather conditions. We took this one for two reasons. First, because it is the simplest type; it may not be the best type of re-circulating system; and secondly because we had been working for 5 or 6 years in a laboratory plant in which we had one large return and the results were criticized, and we wanted to compare the results obtained, so we built the same plant in the research residence in every respect except in length of pipes. We had to vary the length of pipes in order to get the registers in the rooms, otherwise the re-circulation area, the furnace itself, the fuel, the stack heights were the same. We have duplicated that plant in the residence. Our Winter's work is now in report form but will not be available until after the *National Warm Air Heating and Ventilating Association* has issued it.

No. 731

A PROPOSED METHOD FOR COMPARISON OF EFFECTIVENESS OF INDIRECT HEATING SURFACES

By A. E. STACEY, JR. (MEMBER) AND C. M. ASHLEY (NON-MEMBER)

NEWARK, N. J.

THE necessity of a rational method for comparing the merits of different types of heating surfaces has been felt by all interested in the study and uses of indirect heaters.

It is the purpose of this paper to offer a simple system which can be used for comparison between indirect heating surfaces, based on the ratio of Heat Output to Power Input.

Osborne Reynolds in 1874,[1] reporting the results of some experiments on the temperature of air passing through a steam-jacketed boiler tube, pointed out the relationship between the heat transferred and the resistance to the flow of air. In 1895, Dr. T. E. Stanton[2] carried out a series of experiments testing the above relationship. The fluid used in this case was water. Several sizes of steam-jacketed tubes were employed and the tests covered a large range of water velocities. This experiment further demonstrated the presence of a relationship between heat transfer and resistance.

Heat transmission, neglecting radiation, between a surface and a fluid is the result of the diffusion of the fluid near the surface. This may be of two types: one as in viscous flow where the heat is transferred by conduction from one layer in the fluid to the next and the other as in turbulent flow where masses of the fluid from the interior of the fluid are brought near the surface. However, there remains in this case a thin film at the surface in laminar or viscous movement which offers the main resistance to the passage of heat. The heat transfer is much more rapid in the second case than the first due to a greater and more intimate contact between the body of the fluid and the surface.

It is well to note the similarity between this phenomenon and that of resistance to the flow of a fluid through a tube. At low velocities the flow is viscous and the resistance varies as the first power of the velocity; the critical velocity is reached when the parallel motion is broken down and a turbulent motion is introduced. This naturally causes a large increase in the resistance in relation to the velocity

---

[1] Proceedings Manchester Literary and Philosophical Soc., 1874.
[2] Phil. Trans. Royal Soc., Series A. C. K. C.

Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Atlantic City, N. J., June, 1925.

and in the case of simple tubes the exponent of the velocity is approximately 1.82. In the case of ideal impact of a fluid against a surface, the resistance would vary as the square of the velocity.

It has been shown by Coffey and Horne, Carrier and later by Himus and Hinchley that the transmission from a surface parallel to the flow of air is approximately one-half of that from a surface at right angles to it, or in other words the heat transmitted by true impact is twice that of pure parallel flow. On this basis heaters could be divided into three classes, *a*—those with surfaces parallel to the flow of air

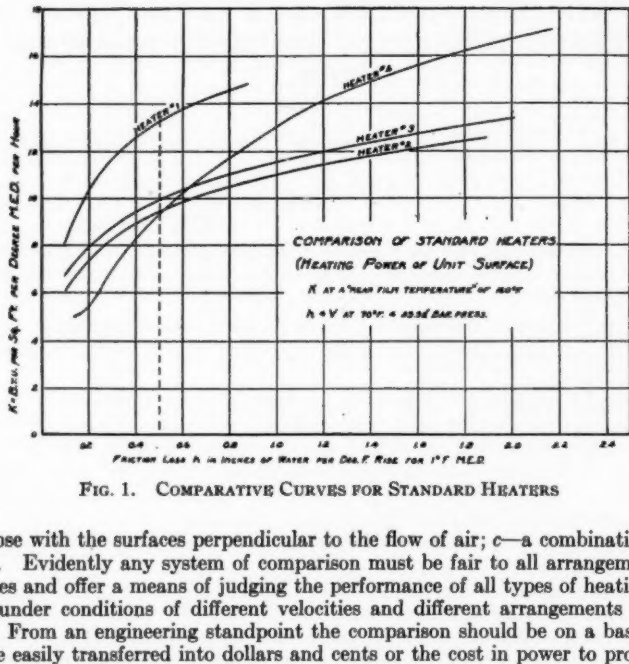


FIG. 1. COMPARATIVE CURVES FOR STANDARD HEATERS

b—those with the surfaces perpendicular to the flow of air; *c*—a combination of *a* and *b*. Evidently any system of comparison must be fair to all arrangements of surfaces and offer a means of judging the performance of all types of heating surfaces under conditions of different velocities and different arrangements of surface. From an engineering standpoint the comparison should be on a basis that can be easily transferred into dollars and cents or the cost in power to produce a definite number of B.t.u.'s under known conditions. It is with these facts in mind that the authors propose the following system of comparison:

Symbols

h = Air resistance loss through heater in inches of water.

V = Air velocity through the *Free Area* of the heater in ft. per min. (corrected to 70 deg. fahr. and 29.92 in. Hg).

A = Free area of heater in sq. ft.

S = Surface of heater in sq. ft.

Q = VA = Air quantity in cu. ft. per min. (corrected to 70 deg. fahr. and 29.92 in. Hg).

C_p = Specific heat of air at constant pressure.
= 0.241 B.t.u. per lb.

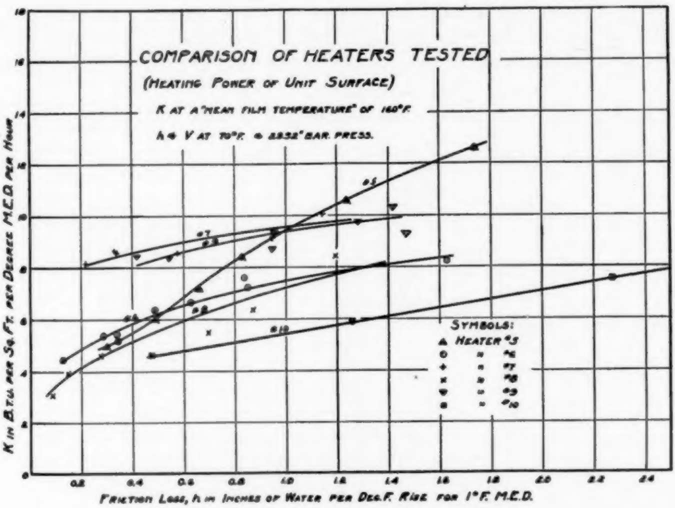


FIG. 2. RESULTS OF TESTS ON SIX HEATERS

D = Density of air at 70 deg. fahr. and 29.92 in. Hg.
= 0.075 lb. per cu. ft.
t_1 = Entering temperature of air in deg. fahr.
t_2 = Leaving temperature of air in deg. fahr.
t_s = Temperature of steam in deg. fahr.
t_f = Mean film temperature of air in deg. fahr.
θ_1 = Initial temperature difference between steam and air. θ_2 = Final, etc.
$M.E.D.$ = Mean effective temperature difference between air and steam (deg. fahr.).
K = Coefficient of heat transmission in B.t.u. per sq. ft. per deg. $M.E.D.$ per hr.
$h \times \frac{M.E.D.}{t_2 - t_1}$ = Resistance loss in inches of water per degree fahr. temperature rise for 1 deg. $M.E.D.$

Basic Equations

$$M.E.D. = \frac{t_2 - t_1}{\log_e \frac{\theta_1}{\theta_2}} \tag{1}$$

$$K = \frac{60Q \times D \times C_p \times (t_2 - t)}{S \times M.E.D.} = \frac{1.085Q \times (t_2 - t)}{S \times M.E.D.} \tag{2}$$

$$= \frac{1.085Q \times \log_e \frac{\theta_1}{\theta_2}}{S} \tag{3}$$

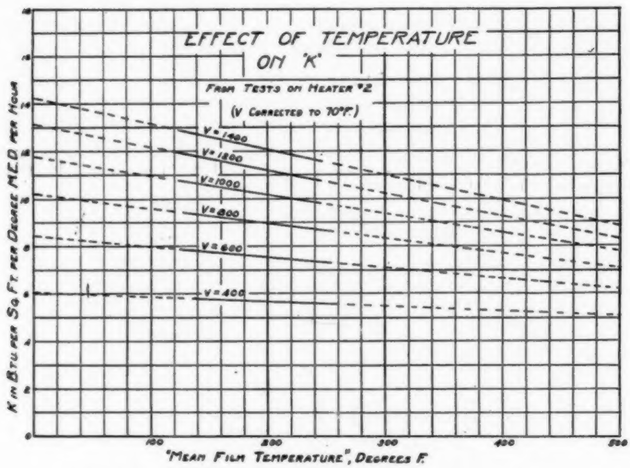


FIG. 3. ILLUSTRATES EFFECT OF TEMPERATURE ON K

$$t_f = t_s - \frac{M.E.D.}{2} \tag{4}$$

$$\frac{V_1}{V_2} = \frac{T_1}{T_2} \tag{5}$$

$$\frac{h_1}{h_2} = \frac{T_1}{T_2} \quad \text{Approx.} \tag{6}$$

The basic problem in making a comparison of indirect heating surfaces is to fix the results obtained from each type so that they will coincide and then compare the amount of surface necessary for each heater to produce the same results. This means that Q, h, t_1, t_2 and t_s, are fixed equal to Q', h', t'_1, t'_2 and t'_s, respectively, and the comparison will be given by the ratio of $\frac{S}{S'}$, for any power input.

Under these conditions (from equation 1) $(M.E.D.) = (M.E.D)'$ and from (2)

$$KS = \frac{1.085Q \times (t_2 - t_1)}{(M.E.D.)} = \frac{1.085Q'\ (t'_2 - t'_1)}{(M.E.D.)'} = K'S'$$

or $KS = K'S'$

that is $\frac{S}{S'} = \frac{K'}{K}$

From which it is seen that $\frac{K'}{K}$ can be used to make the comparison of the heaters with as much validity as $\frac{S}{S'}$.

Now it can be easily shown that for any given type of heater at a given air velocity, h, $\frac{(t_2 - t_1)}{M.E.D.}$ and $\frac{S}{A}$ vary with the depth of heater. Thus $h \times \frac{(M.E.D.)}{(t_2 - t_1)}$ and $\frac{hA}{S}$ are both functions of velocity and type of heater and are independent of depth. Since $Q = VA$ it is possible by altering A to obtain any air quantity desired for a

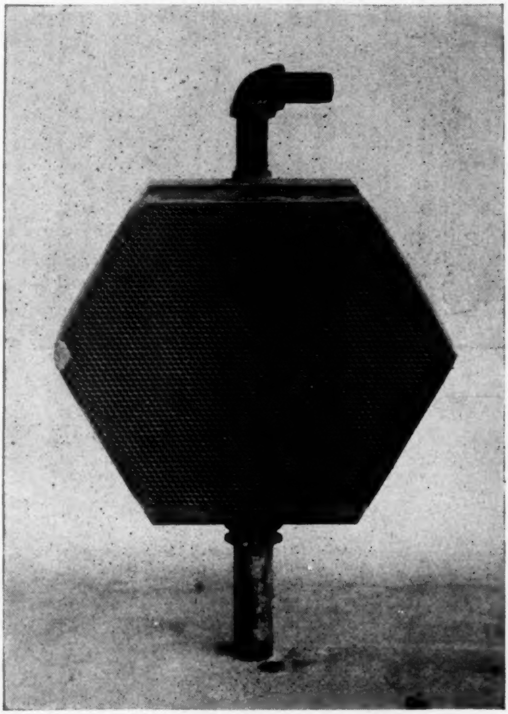


FIG. 4. HEATER NO. 5

fixed velocity without affecting the value of $h \times \frac{M.E.D.}{(t_2 - t_1)}$ or $\frac{hA}{S}$ so long as the depth of heater remains unchanged. Now the expression is in terms of the velocity and type of heater which embodies all the factors it has been desirable to fix. But K is also a function of velocity and type of heater. Thus if K is plotted against $h \times \frac{M.E.D.}{(t_2 - t_1)}$ for the different heaters the ratio of their K's for the same $h \times \frac{M.E.D.}{(t_2 - t_1)}$ gives the desired heater comparison.

In showing that $KS = K'S'$ an expression has been used which involved Q,

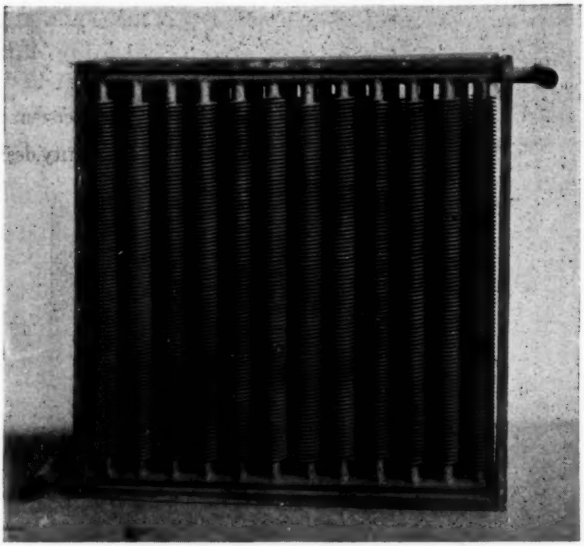


FIG. 5. HEATER NO. 7

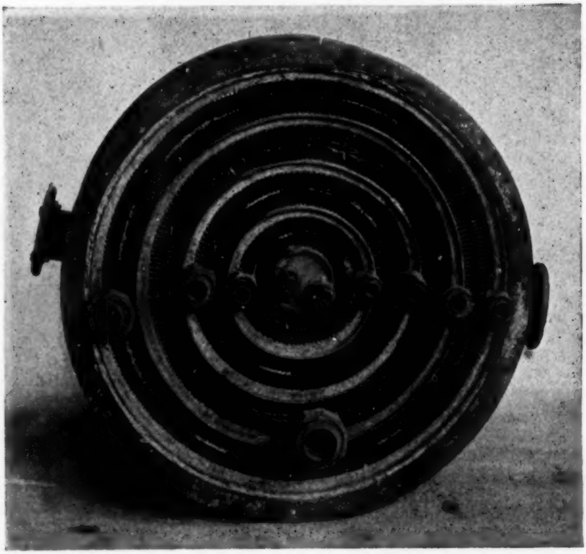


FIG. 6. HEATER NO. 8

t_1, t_2 and t_s. It is apparent that it does not matter whether these terms are individually equal for the two heaters so long as the resulting expression is equal. In order that $\frac{Q(t_2 - t_1)}{M.E.D.} = \frac{Q'(t'_2 - t'_1)}{(M.E.D.)'}$, however, it is necessary that $h = h'$. But it has just been shown that Q can be given any value without affecting either K or $\frac{h \times M.E.D.}{(t_2 - t_1)}$ so long as the velocity is unchanged. If Q is reduced to unity for all cases, knowing that $hQ = h'Q'$ then $h = h'$.

The heaters can now be compared for any conditions of size, shape and velocity.

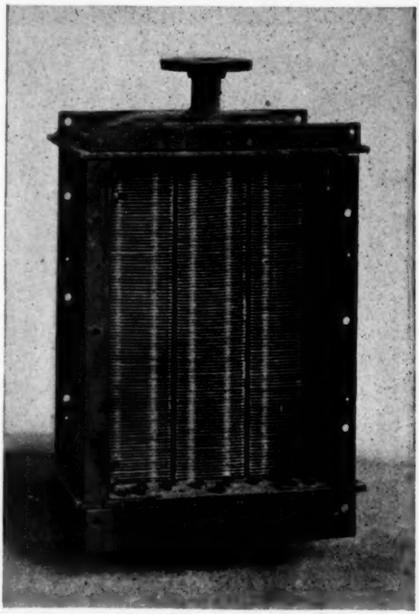


FIG. 7. HEATER NO. 9

It is found, however, that K varies with the *film temperature* (see Fig. 3) and as this variation will depend on the heater used it is best to compare and test heaters at a fixed *film temperature* near the mean of actual operating conditions. Velocity and pressure loss must also be corrected to some fixed temperature which was here chosen as 70 deg. fahr.

Method of Comparison on Cost, Volume and Weight

The heaters can be compared on the basis of cost, volume or weight by dividing K by the cost, volume or weight per square foot of surface and plotting the resultant values which will give comparative curves. In each case the best heater from the

standpoint of the comparison being made is the one whose curve is highest and the heaters compare proportionately to their heights at any value of $\frac{h \times M.E.D.}{t_2 - t_1}$. These curves can also be used as a basis of heater proportioning and design.

Sample Computations for Construction of Curves

Heater No. 3—three row:

$V = 1200$ ft. per min., $t_1 = 60$ deg. fahr., $t_2 = 116$ deg. fahr., $t_s = 227$ deg., $h = 0.230''$ H_2O, $S/A = 52.4$.

Then

$$M.E.D. = \frac{t_2 - t_1}{\log_e \frac{t_s - t_1}{t_s - t_2}} = \frac{116 - 60}{\log_e \frac{(227 - 60)}{(227 - 116)}} = 136.8 \text{ deg. fahr.}$$

$$K = \frac{1.085\ VA(t_2 - t_1)}{S \times M.E.D.} = \frac{1.085 \times 1200 \times 56}{52.4 \times 136.8} = 10.18 \text{ B.t.u. per sq. ft. per degree } M.E.D. \text{ per hr.}$$

$$\frac{h \times M.E.D.}{(t_2 - t_1)} = \frac{0.230 \times 136.8}{56} = 0.561$$

$$t_f = t_s - \frac{M.E.D.}{2} = 227 - 68.4 = 158.6 \text{ deg. fahr.}$$

or approx. 160 deg. fahr.

Sample Comparison

Take on Fig. 1 a value of $\frac{h \times M.E.D.}{t_2 - t_1} = 0.5$ which corresponds with the center of working range of heater No. 1. Then on a comparative surface basis, in terms of heater No. 1, heater No. 2 is 71 per cent as effective, heater No. 3, 75 per cent and heater No. 4, 70 per cent; that is, with equal temperature rise and equal friction, heater No. 2 will require 41 per cent more surface than heater No. 1, heater No. 3, 33$^1/_3$ per cent and heater No. 4, 43 per cent, or stating it in another way heater No. 1 could cost 141 per cent per sq. ft. of the cost of heater No. 2 per sq. ft. and still the resultant heaters would have the same annual charges.

Fig. 1 shows comparative curves of standard heaters, based on data obtained from tables published by the manufacturers of the several distinct types of heaters now being regularly used. These four heaters embody examples of the three types of heating surfaces referred to earlier, that is:

Surface No. 2—Pipe coil heater, transfer from impact.
Surface No. 3—Cast-iron heater, transfer from impact.
Surface No. 4—Simple tube heater, transfer, parallel flow.
Surface No. 1—Gilled tubes heat, transfer by impact and parallel flow.

Fig. 2 gives the results from tests on heaters Nos. 5, 6, 7, 8, 9 and 10.

These tests covered heaters of widely varying characteristics which are shown by the accompanying cuts with the exception of heater No. 5 which had indented tubes. This curve sheet shows the use of this method of comparison as applied to heating surfaces for which no published data is available, necessitating tests in the laboratory.

Fig. 3 illustrates the effect of temperature on K for heater No. 2. It will be noted that at low velocities the variation is very small but at the higher velocities it becomes of such a magnitude as to make necessary the use for comparison purposes of

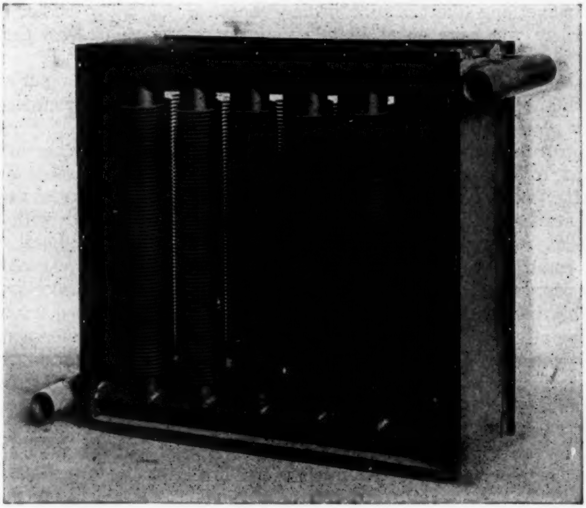


FIG. 8. HEATER NO. 10

a constant *film temperature*. This has been done in the present comparison t_f being taken 160 deg. fahr. approx.

Summary

First. The ratio between K in B.t.u. per sq. ft. per degree per hr. and friction in inches of water per deg. rise for 1 deg. fahr. *M.E.D.* has been proven to be a proper basis of comparison between heating surfaces.

Second. Examples of the use of this system of comparison between heating surfaces have been shown both for heaters where complete data is available and also for heaters for which there are only a few tests.

DISCUSSION

W. H. CARRIER: Mr. President. The principal thing in any heater comparison, or in considering any heater performance, is that, roughly, for an increase of five times in the pressure, that is, if you increase your pressure five times across

your heater you will get about twice the heat transfer. Keep that figure in mind—five times the pressure increase will give you, roughly, twice the amount of heat transfer. There is absolutely no way in which you can compare heaters with velocities. Velocity doesn't mean anything. Take a heater that is of unusual shape, take the Vento, for example, or even a pipe coil, and you have airways varying all the way through the heater. The velocity at the point of minimum contraction doesn't mean an awful lot, because effective velocities are often at points of considerably greater area. Velocity doesn't mean anything unless it be a plate type of heater where there is an absolutely uniform distance, or unless you have something like the tube heater, the U. S. Cartridge heater or similar designs, where you have an absolutely uniform cross-section, then the velocity would have a real meaning, but the majority of heaters do not have any such uniform airways, and it is absolutely impossible to make a comparison of one heater with another. In such cases if they are all tube heaters you can compare them very nicely with the velocity because the velocity would have a definite relationship to the resistance. For that reason it was necessary, as Mr. Stacey pointed out, to have a basis of comparison which shall be independent of velocity. The velocity loses its significance entirely when comparing heaters of varying and diverse character. It is very common to say, "Here is a heater with a certain velocity that will give you twice the transmission of another heater." Well, when you come to analyze it, it also has, perhaps, four times the resistance for the same velocity, or perhaps five times the resistance for the same velocity, and you could simply take the other heater and raise the velocity and you would get the same temperature difference. There would be absolutely no difference between them. So you see it is absolutely impossible to compare heaters on velocity as we have ordinarily done. It is absolutely fallacious. I am not saying this to make a comparison unfavorable or favorable to any heater, but it is simply the fact. For example, you take the tube heater and you must have a considerably higher velocity to get the same heat transfer, but that doesn't mean that it is a poorer heater. Perhaps it is a much better heater.

E. W. LEGIER: Probably one of the most important points in applying a practical application of a heater is noise, particularly with regard to public building work. I would be interested to know if Mr. Stacey in his investigations determined anything different than is given in the issue of the code with regard to critical velocities insofar as noise is concerned.

In regard to the critical velocities, we have shown in one of those heaters the point where the critical velocity is reached. That is the heater which has the curve that reverses at the lower point. The critical velocity through most heaters is very low, in fact, lower than the tables are brought to.

E. H. LOCKWOOD (written): The proposed method for comparison of indirect heating surfaces will be read with interest by experimenters with such equipment. The principle advanced by the authors, and clearly stated in the first paragraph of the summary, is undoubtedly correct. It is not necessary, however, to use the author's notation in order to apply the principle. The writer has used what seems to be the same method in a different form in making comparisons of various indirect heaters. The method has been to determine for each type of heater the number of stacks, size, etc., needed to produce the same heating output in each case.

It requires that each heater shall work under the same steam pressure, same initial and final air temperatures, and shall handle the same number of cubic feet of

air per minute with the total air friction not exceeding a fixed maximum. Subject to these conditions each heater can use any desired air velocity, or free air area, number of stacks and extent of surface. It is usually possible to make such a comparison of different heaters, using only data from the manufacturer's catalog. It may be of interest to see the writer's version of such a comparison for two widely different types. The following figures were interpolated from catalog tables and are believed to be substantially correct, although they are offered as illustrative rather than as strictly exact calculations.

COMPARISON OF INDIRECT AIR HEATERS

Volume, temperature and friction requirements:

Vol. of air to be heated, c.f.m	3600
Initial air temperature, deg., F	0
Final air temperature, deg. F	80
Steam pressure by gage	5.0
Air friction through heater, in	0.20

Dimensions of cast iron heaters from catalog data:

The specified air friction can be had by 1 stack at air velocity of 1800. f.p.m. 2 stacks 1350 f.p.m, or 3 stacks at 1150 f.p.m. The temperature table shows that three stacks will heat air to 82 degrees, hence 3 stacks will be chosen. The free air area through the heater works out $3600 \div 1150 = 3.13$ sq. ft. The next largest catalog size is 3.22 sq. ft., which will be chosen as free air area. From the radiator dimensions the following figures are obtained: Frontal area 7.8 sq. ft. Depth of three stacks 28 in. Heating surface 168 sq. ft. Space occupied 18.2 cu. ft. Weight 1375 lb.

Dimensions of copper heaters from catalog data:

The specified air friction can be had by 1–6 in. stack at 1600 f.p.m. air velocity, or 1–5 in. stack at 1530 f.p.m. The temperature table shows that the 6 in. stack will heat the air to 83 deg. at the velocity stated, hence this stack will be used. The free air area figures $3600 \div 1600 = 2.26$ sq. ft. From the catalog the remaining dimensions are found to be: Frontal area 3.60 sq. ft. Depth 6 in. Surface 191 sq. ft. Space occupied 1.8 sq. ft. Weight 131 lb.

If a comparison is based solely on surface the cast iron type will be judged more effective, since 168 sq. ft. of cast iron surface accomplishes as much as 191 sq. ft. of copper surface. If a comparison is based on volume or weight, the advantage is largely with the copper type which has only one-tenth the volume and one-tenth the weight.

While the above comparison does not appear to follow the rule advocated by the authors of the paper, yet it does so in substance since the same M. E. D., h and (t_2-t_1) are used for each heater. It is evident that the surface comparison is by no means the most important in extreme cases, where the heaters differ so widely in size and weight.

In the illustrative example the copper heater was shown to have slightly less effective surface than cast iron for the conditions of the problem. It should not be assumed that this is true for all cases, nor for all types of brass and copper heaters. The cellular type of heaters differ considerably from each other, and a few are superior to any form of cast iron heaters in effectiveness of surface, as well as in volume and weight.

COMPARATIVE TABLE OF DIMENSIONS FOR BOTH HEATERS

Type of heater	Cast iron	Copper
Free air area, sq. ft.	3.22	2.26
Vel. through free air area, f.p.m.	1150	1600
Number of stacks required	3	1
Depth of each stack, in.	9 1/8	6
Total depth of heater, in.	28	6
Frontal area, sq. ft.	7.8	3.6
Space occupied, cu. ft.	18.2	1.8
Surface, total, sq. ft.	168	191
Weight of heaters, lb	1375	131

No. 732

DRY AIR FILTERS

By S. E. Dibble, Pittsburgh, Pa.

MEMBER

DURING the past six months the writer has done some very interesting work in testing a number of dry air filters at the Carnegie Institute of Technology. These tests prove that considerably more may be expected from this type of filter in the future than is realized at the present time.

While making these tests, the first thing which presented itself was the method to be used in testing. For the future development of these filters, it is important that the Society adopt a code for testing air filters, so that there will be a basic method upon which all filters can be tested.

The tests were made in a room, 10 x 15 x 9 ft. Three ducts were attached, running to a large main duct, which, in turn, was connected with a fan. A filter unit was located where these ducts entered the room, one instrument was placed in front of the filter and a duplicate instrument in back of the filter in the duct. Dust was liberated in the room by means of a dust liberator, which discharged an even quantity of dust at all times during the test. The instruments which were used for testing the efficiency of these filters were: The Cottrell Dust Precipitator; the A-A Dust Determinator; and the Hill Dust Counter.

The most important function of the filter is to remove the dust from air which passes through it. This is the vital function of all filters and should be the controlling factor. Another important item is the time element. By this is meant the amount of time it takes before a filter is filled with dust and ceases to function. During the tests, it was discovered that certain types of filters would continue to remove dust with a high efficiency considerably longer than others, which indicated, of course, that this type of filter would not have to be cleaned so often, and would have a much lower maintenance charge in the course of a year. Still another feature which is of importance is the power which is required to draw the proper amount of air through the filter. As a filter cleans air it plugs or stops the air passages, making them much smaller, and, as a result, more power is required to deliver the original amount of air.

Another feature of this type of filter, which was very prominent, was the way in which dust gathers on all surfaces of the filtering medium. The dust sticking to the surfaces which are coated with oil gradually becomes covered with oil and acts as a dust collector. In some types of filters this piling up of dust increases the

Presented at the Semi-Annual Meeting of the American Society of Heating and Ventilating Engineers, Atlantic City, N. J., June, 1925.

total surface which is exposed to the passing air and instead of decreasing the efficiency of the filter as the filter becomes clogged, it increases the efficiency up to a certain point. As the oil surfaces collect dust, they gradually build up and block or fill all the air passages. In doing this, the volume of air which it is possible to pass through the filter is gradually reduced and the velocity of the air passing through is increased. If the volume of air is reduced, it means an unbalanced system with an ever reducing quantity of air. If the velocity is increased, it means the lowering in efficiency of the filter because the dust will slide off and plug the openings, and in some cases the impact is great enough to overcome the tenacity of the oil on the surfaces of the metal.

All the dust cannot be removed by these filters, and a 100% efficient filter, of course, means the removal of all the dust. Therefore, the writer believes that the Society should undertake the problem which confronts this industry, that is, determine the point at which filters cease to function, and the kind and type of dust for which these filters are best suited. If established, this would represent 100% efficiency, and could be used as a standard.

During the tests which have been running, the Cottrell Dust Precipitator as representing 100% dust removal was used, results showed that filters which were tested with this instrument have an efficiency of around 60%. The same filters tested with the A-A Dust Determinator have an efficiency of about 80%, and the same filters tested with the Hill Dust Counter will have an efficiency of around 95%—all of these filters being tested under the same conditions and with the same dust.

These tests point out the advisability of deciding which instruments shall be used in making comparative tests on filters and which instruments shall be used to represent the efficiency of filters. Until this is done, there is going to be a great discrepancy between the results published by the various manufacturers, which will mean an unsettled condition for this industry.

From the tests conducted, the writer has arrived at two conclusions regarding filters: First, as the pressure is applied in the filter, and this pressure shows a continued increase, the efficiency rises; Second, when the filter builds up a resistance to a certain point and then stops increasing the efficiency drops.

DISCUSSION

J. D. CASSELL: May I ask Prof. Dibble if he found in his tests that as the orifices closed up with dust he got a larger percentage of dust compared with the amount of air going through?

PERRY WEST: I should like to ask in connection with the Cottrell instrument if the quantity of dust coming through is measured by weight or the number of particles.

F. R. STILL: Mr. Chairman, I have great faith in the ultimate perfection of the dust determining machine the Laboratory has been developing. It may never be perfected to the point where it will catch all of the dust there is in the air, but, to my mind, it is unnecessary to accomplish this for the purpose the machine is intended.

This machine works on the principle that very dusty or dirty air will completely clog the filter in a certain period of time. When it becomes impossible to get any more air through the filter, the resulting condition represents 100% dirty condition of the air.

On the other hand, when a sample of air has been thoroughly cleaned before passing it through the machine, then of course the machine will collect nothing on the filter in the same period of time and there will be no rise in pressure above the normal pressure required to pass the air through the clean filter.

These two conditions represent the two extremes of measurement, and any resistance to the flow of air, all of which can be plotted on a chart to represent certain percentages for immediate comparison.

The portability and the ease by which comparisons can be made by the laboratory machine is particularly attractive to me. Nobody minds how many particles of dust there may be in the air, but we do want to know what condition is harmless, and we want a machine which can measure the relative amount of dust so as to determine whether, after the air has passed through an air washer or filter, the dirt has been removed to a point where the air is harmless. We also need some one standard for the purpose of comparison. As the Laboratory has spent so much time in developing this machine, I think it should proceed to perfect it, and when it has reached this point it should be adopted as the standard of this Society.

THORNTON LEWIS: I would like to ask if Mr. Dibble made any comparative tests of filters and air washers and what these tests showed.

J. I. LYLE: I have had some experience with the Cottrell instrument and I don't think there is any question but that it is very much more efficient than any of the other measuring devices that we have had, but it's quite an instrument, to say the least. It is expensive and rather difficult to handle; it is almost impossible to use it on any commercial installation.

If we can once establish that any one instrument has a constant percentage of efficiency we will have accomplished a great deal. It doesn't make much difference to us what that percentage is, whether it is 70%, 80% or 90%, so long as it is constant. I would like to ask Prof. Dibble if in his comparison of the various instruments he has found that there is a constant ratio between the efficiency shown, say, by the A-A Dust Determinator and the electric precipitator. Knowing that efficiency, we could use the A-A instrument or the Hill instrument and get our percentage without any trouble.

We have had some experience in the last year in the testing of filters. Our tests were made rather with the idea of determining for our own information the relative efficiencies of different makes of filters, and we have found, at least the indications are, that the efficiency of the filters falls off very fast when they become dirty, or, in other words, when they fill up sufficiently to require a high velocity. The members of our organization, however, differ quite materially as to which ones of the various filters that we tested were the best. For instance, some of them collected the dirt almost entirely on the face, others collected it all the way through the four inch thickness. With the ones that collected the dirt on the face, of course, the resistance increased faster. They seemed to have less resistance in the beginning and greater resistance with a given amount of dirt or a given amount of time. Some of us took the view that the ones who collected it all the way through, although they didn't collect quite so much, wouldn't need cleaning so often. Others of us felt that those who collected on the face were better because you could see them when it was dirty.

DEAN F. PAUL ANDERSON: There was an attempt made by the Laboratory to develop a method of measuring the dust in the atmosphere which would give a commercial instrument to the engineering profession.

We think that has been accomplished, because the A-A Determinator gathers all the dust and you can do what you please with it. It is a sort of instrument that can be carried around and can be operated by any intelligent operating engineer. This whole question of dust is like many other subjects coming before this great Society.

All information is valuable and each investigator's work contributes to the general understanding of the scientific principles involved in any engineering question.

PRESIDENT DIBBLE: As the air passageways fill up with dust we found that the dust itself becomes coated with oil, and the efficiency of the filter gradually builds up. That is, you have a larger oil surface exposed for the passage of air as the filter builds up with oil and dust. Instead of having a flat surface in front, or flat surface throughout, you have a little rounded surface built up with dust in each case and the oil gradually soaks around each particle of dust. In checking the Cottrell instrument, the dust is all measured by weight. I agree with Mr. Lyle that it takes a very good man to operate the instrument and see that it is handled properly. I have taken the Cottrell Percipitator as meaning pretty nearly 100%. I can take cigarette smoke and blow it through the small instrument we have and it is stopped and if it stops cigarette smoke it stops a good deal of other dust in the air that is objectionable. I made no comparison between washers and dry air filters. I have found that dust, especially the heavy particles, collect on the front surface and the finer particles collect throughout the entire body of the filter, and therein lies, I think, a very important point in the construction of filters.

J. I. LYLE: Did you make any determination to find out whether various instruments seemed to be constant as compared with the Cottrell instrument?

PRESIDENT DIBBLE: Yes, by looking over our records we find that the A-A keeps a fairly even line. That is, it is more even than the other.

No. 733

HOW DUSTY IS AIR?

By Margaret Ingels, Pittsburgh, Pa.

MEMBER

THE purpose of the investigations that are presented in this report is to learn what the dust problem is, that is, to learn how much dust is contained in air in various types of rooms.

The A-A Dust Determinator was the instrument used to measure the dust, and the tests were made as follows: The instrument was placed in the room where it would get the most representative sample of dust. Air was sampled at the rate of 0.4 c.f.m. over a period of one-half hour. Each sample totaled 12 cubic feet. The resistance across the filter medium was recorded in inches of water every five minutes of the test and the resistance plotted against time. If the dust content of the air were uniform throughout the test the points plotted would be on a straight line. If there were a variation of the dust content the average was determined by drawing a straight line average through the points. The initial and final resistances thus obtained were used to read the amount of dust in the air. The amount was read in A-A Dust Units from the Dust Chart.[1]

A series of tests was made in each room over a period of time varying from eight hours to three days. One test was made every hour, usually starting on the hour.

Table No. 1, gives the data collected showing the types of rooms and their characteristics, the maximum and minimum dust contents and the average dust content in A-A Dust Units.

There is one fact about this data that should be thoroughly understood. Where two rooms of like use are reported, one having "forced ventilation" and the other "natural ventilation" the difference of dust content does not indicate the cleaning efficiency of the ventilating equipment. In most instances the forced ventilation equipment was necessary due to the extreme conditions not found where the natural ventilation is still used.

There was a decided difference in the quality of dust in some of the rooms tested, and this difference in quality was due to the sizes of the dust particles. In the column of Table No. 1 headed "Remarks," the condition of the air according to the quality of dust it contains is designated as "naturally dusty air" or "artificially dusty air." "Naturally dusty air" is that which contains such small dust particles that they stay suspended over a long period of time. "Artificially dusty air" is that which contains such large particles of dust that they settle out quickly.

Naturally dusty air is that found in most public buildings, is that which the

[1] "The A-A Dust Determinator," Margaret Ingels, Jan. Journal, 1925.

TABLE I

Type of Building	Room	Type Ventilation	Maximum Dust A-A's	Minimum Dust A-A's	Average Dust A-A's	App. Wt. of Dust in Grams per Cu. Ft.	Remarks
Dept. Stores	Balcony or first floor						
123 tests	Store A	Natural	.933	.200	.446	.000014	Naturally dusty air
	Store B	Natural	.942	.275	.411	.000013	Naturally dusty air
	Store C	Natural	.900	.392	.608	.000019	Naturally dusty air
	Store D	Natural	1.083	.233	.586	.000018	Naturally dusty air
	Basement						
	Store A	Washer	.375	.233	.299	.000009	Naturally dusty air
	Store B	Washer	.375	.200	.307	.000010	Naturally dusty air
	Store C	Washer	.563	.333	.464	.000015	Naturally dusty air
	Receiving Room						
	Store B	Natural	.650	.360	.485	.000016	Naturally dusty air
	Store C	Natural	1.075	.300	.572	.000018	Naturally dusty air
	Store D	Exhaust Fans	.483	.317	.392	.000013	Naturally dusty air
	Rug Department						
	Store B	Natural	.500	.333	.416	.000013	Naturally dusty air
	Store C	Natural	.750	.333	.459	.000015	Naturally dusty air
	Store D	Natural	.960	.733	.856	.000028	Naturally dusty air
	Shipping Room						
	Store B	Natural	.492	.417	.450	.000014	Naturally dusty air
	Second Floor						
	Store A	Natural	.350	.200	.270	.000008	Naturally dusty air
	China Packing Room						
	Store D	Natural	.967	.417	.612	.000053	Artificially dusty air
Hotels	Lobby						
59 tests	Hotel A	Natural	.937	.500	.687	.000022	Naturally dusty air
	Hotel B	Natural	.659	.433	.517	.000017	Naturally dusty air
	Hotel C	Natural	1.194	.317	.785	.000026	Naturally dusty air
	Hotel D	Fan	1.333	.479	.783	.000026	Naturally dusty air
	Hotel E	Air Washer	.691	.416	.480	.000016	Naturally dusty air
	Hotel F	Natural	1.050	.709	.858	.000028	Naturally dusty air

Table I. (Concluded)

Type of Building	Room	Type Ventilation	Maximum Dust A-A's	Minimum Dust A-A's	Average Dust A-A's	App. Wt. of Dust in Grams per Cu. Ft.	Remarks
Schools	Boys' Play Room						
81 tests	School A	Natural	.73	.23	.436	.000039	Artificially dusty air
	Kindergarten						
	School A	Split System	.72	.25	.476	.000016	Naturally dusty air
	School B	Semi-dry	.44	.192	.278	.000009	Naturally dusty air
	Application Room						
	School A	Split System	.42	.25	.31	.000010	Naturally dusty air
	Main Hall						
	School A	Split System	.82	.32	.482	.000016	Naturally dusty air
	School B	Semi-dry	.42	.17	.310	.000010	Naturally dusty air
	Class Room						
	School A	Split System	.90	.21	.400	.000013	Naturally dusty air
	School B	Semi-dry	.25	.17	.192	.000006	Naturally dusty air
	School C	Washer	.36	.10	.256	.000008	Naturally dusty air
Railroad Terminals	Main Waiting Room						
36 tests	Station A	Washer	.867	.300	.583	.000018	Naturally dusty air
	Station B	Natural	.858	.200	.384	.000012	Naturally dusty air
Steel Mills	Bending and Welding						
19 tests	Floor						
	Plant A	Natural	1.55	.60	1.091	.000088	Artificially dusty air
	Plant A	Natural	.750	.25	.386	.000034	Artificially dusty air
Industries	Power House						
24 tests	Plant B	Natural	.250	.167	.230	.000020	Artificially dusty air
	Enameling Room	Natural	1.375	.354	.784	.000070	Artificially dusty air
	Charging Platform	Natural	.708	.292	.446	.000040	Artificially dusty air
	Foundry	Natural	1.313	.867	1.030	.000076	Artificially dusty air

ventilating engineer is more often required to clean, and it is decidedly the harder to clean. That it is objectionable from the health standpoint is not yet known, but it is disagreeable because it discolors.

Artificially dusty air is that found in most industries or where the activities in the rooms introduce dust into the air. It is usually objectionable from a health standpoint, and because it is often visible it is undesirable due to the psychological effect on the occupants of the room. It is the easier dust to remove and the air is clean soon after the activities which introduced the dust have ceased.

In the column of Table No. 1 headed "Approximate Weights of Dust," the values given are not absolute. They are included in the table for comparative

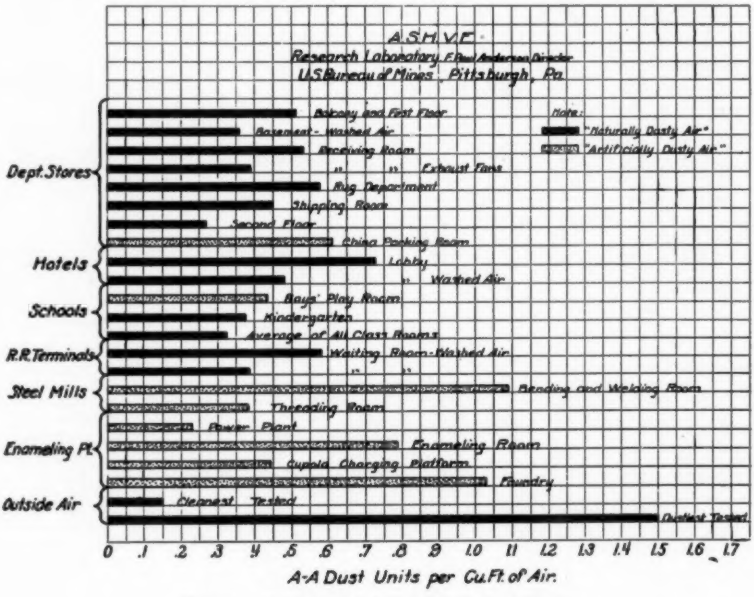


FIG. 1. RECORD OF DUST DETERMINATIONS

values only. In Fig. 1 the amounts of dusts in A-A Units obtained in these tests are shown graphically. The two qualities of the dusts found are distinguished by the solid lines and the stippled lines.

The cleanest outside air tested in Pittsburgh and the dustiest outside air tested are shown on the chart.

It is desirable to have clean air but the degree of cleanliness necessary for health and comfort has not been fixed. The data herein compiled show the amounts of dusts in the various public buildings and industrial plants which have been tested to date.

The data include that collected by Miss Helen Cochran, Secretary of the Research Laboratory and C. A. Herbert, Assistant Research Worker of the Laboratory. The writer wishes to acknowledge their careful and painstaking investigations, and their interest throughout the work.

In Memoriam

Names	Joined the Society	Died
John Black	June 1919	July 1925
Max Bodtke	Dec. 1921	Mar. 1925
J. R. Furman	June 1919	Sept. 1925
C. S. Haire	Sept. 1920	Feb. 1925
John Long	Sept. 1916	May 1925
E. C. Molby	July 1915	Mar. 1925
Abram C. Mott, Sr.	1897	May 1925
William A. Pittsford	June 1919	Apr. 1925
J. G. Walsh	Mar. 1919	Aug. 1925

Abram Cox Mott, Sr.

The death of Abram Cox Mott, Sr., chairman of the board of directors of the Abram Cox Stove Co., Philadelphia, Pa., occurred on May 6, 1925. The death of Mr. Mott at the age of 75 years, marks the passing of one of the most widely known and highly respected members of the heating industry and his passing is regretted by his fellow members in the Society. Having entered this line at the age of 16, he devoted the remaining 59 years of his life to building up the firm now known as the Abram Cox Stove Co.

Mr. Mott was born in 1850 at Glen Falls, N. Y. In 1866 he came to Philadelphia and obtained a position as molder's apprentice in the foundry of the Cox, Whiteman & Cox Co. His two uncles, Abram and Joseph Cox, were partners in this firm, which had been established in 1847. Mr. Mott worked at successive jobs for this company for seven years until in 1873 he was made general superintendent. In 1881, upon the death of Whiteman and Joseph Cox, Mr. Mott bought out their interests and since 1882 the firm has been known as the Abram Cox Stove Co.

Abram Cox was president of this company from 1882 until he died in 1885, but was unable to take an active part in the business because of ill health. During these three years, Mr. Mott, in the capacity of assistant to the president, guided the destinies of the company, and under his management its business was greatly increased.

In 1885 Mr. Mott became president of the Abram Cox Stove Co., which marked the beginning of an era of great progress for the company. He served as president until 1924, taking an active part in the business until 1922, when his son, A. C. Mott, Jr., assumed control of the company a svice-president. In 1924 Mr. Mott was elected chairman of the Board of Directors, and his son became president. Mr. Mott survived four different changes in the officers of the company, and remained at the head of the business from 1882 until the time of his death, a period of 43 years. In 1901 the Thomas, Roberts, Stevenson Co. was purchased, and in 1917 it was merged into the Abram Cox Stove Co. He was familiar with every phase of the heating and cooking appliance business and was responsible for the invention of several appliances including the Novelty Circulator, one of the first cast-iron heating boilers.

Mr. Mott's hobby was agriculture and horticulture and he spent much of his time directing the operation of his large farm near Philadelphia. Mr. Mott was a member of the Union League, Philadelphia Country, Manufacturers, and Edgemere Country Clubs. He was one of the oldest members of the Society having been elected in 1896 so that he might well be considered one of its founders. He had seen it grow practically from an idea and served it well during its progress of the last 30 years. His life was spent in the heating industry and he contributed much toward its progress.

He leaves a widow, two sons, A. C. Mott, Jr., president of the Abram Cox Stove Co., and Merle Mott, secretary of the company; one daughter, Mrs. Green, all of Philadelphia; one sister, Mrs, Alice M. Hazlett of California; and one brother Edwin P. Mott, treasurer of the Carr Supply Co., Chicago.

INDEX